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DESIGN OF RECIPROCATING SINGLE CYLINDER EXPANDERS FOR RANKINE CYCLE ENGINES

R. D. Brooks

General Electric Company

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Environmental Protection Agency

4 January 1972

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QUARTERLY PROGRESS REPORT

DESIGN OF RECIPROCATING SINGLE CYLINDER EXPANDERS FOR RANKINE CYCLE ENGINES

ENVIRONMENTAL PROTECTION AGENCY
CONTRACT NO. 68-01-0408
PROGRAM MANAGER–R. D. BROOKS

October 4, 1971 to January 4, 1972

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Details of illustrations in
this document may be better
studied on microfiche

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I

DESIGN OF RECIPROCATING SINGLE CYLINDER
EXPANDERS FOR RANKINE CYCLE ENGINES

Quarterly Progress Report
October 4, 1971 to January 4, 1972

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1.0 SUMMARY

The Environmental Protection Agency is sponsoring development programs to identify automobile engines having low exhaust emissions. One approach being investigated is the Rankine cycle which includes a reciprocating steam expander as the prime mover. For maximum engine performance, high temperatures and pressures similar to modern steam power plant practice are required. For these operating conditions, liquid lubricants for cylinder and inlet valve lubrication are a limitation. Solid lubricants, to be investigated in this program, appear to offer long life at high temperatures. Potential system contamination problems from liquid lubricants introduced to the working fluid are also reduced.

Phase I of this program includes the design of practical single cylinder reciprocating expanders for the purpose of evaluating solid lubricants and other supporting materials for use in a Rankine cycle automotive propulsion system utilizing a water base fluid. The Phase I investigation has the following objectives:

- Establish preliminary designs of trunk piston and crosshead piston single cylinder expanders.
- Evaluate expander materials technology for higher temperature operation with solid lubricants.
- Evaluate solid and liquid lubricants for reciprocating expanders.
- Build and test one trunk piston expander and one crosshead piston expander at conditions representative of the engine system.

Phase I offers the opportunity to conduct studies of expander design and materials trade-offs utilizing selected advanced technology that will be appropriate for the automotive application. The program is comprised of a four task effort arranged and scheduled to adequately compare the two expander types using similar materials of construction. The efforts of the first quarter are summarized in this report and include the following results:

1. The technology base has been established for trunk piston and crosshead piston designs supporting materials technology, and lubrication state of the art.
2. Design tradeoff studies have been completed to arrive at preliminary designs of a single cylinder, trunk piston expander and crosshead piston expander.
3. A selection was made of constructing materials applicable to operation at inlet conditions of 1000°F and 1000 psia with solid lubricants.
4. Lubrication studies were conducted to identify suitable approaches for each design including selection of solid and liquid lubricants.
5. Manufacturing drawings were completed and issued for the crosshead piston engine. Procurement of all parts was initiated. Test facility design was completed and construction has begun.

2.0 TASK I EXPANDER DESIGN

2.1 Expander Design Technology

In conformity with contract guidelines, the general reciprocating expander configuration was initially selected to be that of a single acting, simple expansion engine. This selection is in accordance with the General Electric proposal, and is supported by the conclusions of other EPA funded studies. Two single cylinder expanders are to be designed, built, and tested, one a crosshead piston type and the other a trunk piston type. Steam inlet conditions are 1000°F, 1000 psi. The exhaust pressure is 20 psi. These expanders are to be representative of one cylinder of a four cylinder engine of approximately 150 hp output, suitable for automotive application. A governing requirement of these designs is that solid lubricants be employed for the piston rings, piston rod packing (where employed), and valve stem seals. Such use of solid lubricants is intended to implement the development of an engine free of lubricating oil contamination in the steam.

The task of establishing the technology base for the design of the two expanders was handled by making the following investigations:

1. High speed steam engine valving configurations, and valve gear designs.
2. Wrist pin bearing design suitable for high unidirectional loading.
3. Oil free high temperature piston ring design.
4. Oil free valve stem sealing, friction, and wear.
5. Piston rod sealing.
6. General reciprocating engine mechanical design.

In close coordination with these investigations corresponding materials selection investigations were carried on. These are described in Section 3.0.

Salient results and conclusions of these investigations are briefly discussed below.

2.1.1 High Speed Steam Engine Valving Configurations and Valve Gear Designs

2.1.1.1 General Valving Configurations for High Speed Steam Engines

Valving configurations which were identified as being of interest for preliminary design studies include the following:

1. Uniflow engine with spring loaded recompression relief exhaust valve, simple exhaust porting and engine actuated inlet valve, either in the cylinder head or in a side steam inlet passage.
2. "Zero" clearance volume engine with exhaust valve and inlet valve in the cylinder head.

Valve types of interest include poppet valves of several varieties, sleeve valves, and rotary valves. Sleeve and rotary valves minimize or eliminate the high pressure loading on the actuating mechanism. Rotary valves in particular are of interest because they offer potential capability for a simple variable cut-off mechanism. However, because of the formidable sealing problems of these valve types, it was decided that such an approach involved too much technical risk for the presently contracted program with its very tight schedule. Investigative effort was concentrated on unbalanced and partially balanced cam actuated poppet valves. The principal design problems identified were that of high Hertz stress between cam and cam follower (resulting from the 1000 psi steam pressure against which the valve must be opened) for the unbalanced poppet valve approach, and that of steam leakage for the various types of partially balanced poppet valves. It was recognized that valve train accelerations are quite high because the cam shaft must operate at engine speed (as opposed to half engine speed for four cycle internal combustion engines). However, it was found that the valve train inertia forces, at least for the case of the unbalanced poppet valve, are dominated by the high steam pressure loading forces. This fact, although it results in the high Hertz stress problem, simplifies the dynamic design of the valve train.

2.1.1.2 Variable Cut Off Mechanisms

The expander load control mode for the four cylinder engine will be that of variable inlet valve cut off which must cover, at a fixed speed, a range from zero steam admission to a maximum value determined by the limit of the steam supply. The status of high speed variable cut off technology was reviewed, and the following possible alternative design approaches were identified:

1. The Caprotti mechanism, which employs a single poppet valve actuated by a yoke bearing on two variably phased cams. This mechanism is successfully employed in Skinner steam engines. For a high speed application requiring a wide range of cut off, very high valve train accelerations are involved.
2. Hydraulically operated poppet valve. A hydraulic valve actuator operating on the principle of the Bosch injector has been successfully employed for achieving variable valve period by The Waukesha Motor Company. For a high speed steam expander requiring a camshaft running at full engine speed, however, it appears doubtful that such a system has a fast enough response.
3. A variable fulcrum type of overhead valve actuation mechanism. Such a mechanism has been invented and developed by the Ford Motor Company. Preliminary design studies of several forms of this device were made (see below) with the cooperation of the Ford Motor Company. Various problems relating to overall expander height, and lubrication of the variable fulcrum mechanism resulted in a decision not to pursue this approach. A further complication is the fact that this approach to variable cut off requires a mechanism to vary the phase of the cam relative to the crankshaft.
4. A rotary valve mechanism utilizing a rotatable cut off sleeve between the rotating valve and the valve housing. As mentioned above, the formidable sealing problems of this approach do not appear to be solvable without extensive development effort not permitted by the schedule of the current contract.

5. Two poppet valves in series, variably phased to provide a change in the period of overlap between openings of the two valves. Advantages of this approach include the following:

   a. Long cam event angles can be employed. This minimizes the problems of cam design, permitting reasonable acceleration, high lift, and reasonably small cam base circle diameter.
   b. Full cut off range (zero to any desired maximum) can be readily achieved.

   Problems are also recognized for this approach as follows:

   a. Because of the flow resistance of two valves in series the valves must be larger or employ higher lift than if a single valve is used. Large valves result in high valve train loads unless balanced valves are used. Very high lift tends to require long cam events, large cam base diameter, high cam rubbing velocity and high valve train acceleration.
   b. With two series valves, it is difficult to realize the designed 5% clearance volume in a uniflow type expander.

   Two alternate cam phasing mechanisms were identified, one involving the use of helical splines, and the other involving a planetary gear train between crankshaft and camshaft, with phasing variation achieved by rotating the planet carrier.

After evaluating the above alternate approaches to variable cut off design, it was decided to accept the two series poppet valves as the ultimate objective approach. For the single cylinder test engines, it was decided to employ a single poppet type, fixed cut off valve gear. It is believed that this approach will provide a test of all the "basic barrier" features of poppet type valving (principally that of high valve train pressure loads), and will permit the later introduction of the two series valve type of variable cut off mechanism as a routine design modification.

2.1.2 Wrist Pin Bearing Design for High Unidirectional Loading

Because of the high MEP of the steam expander under long cut off conditions, and the fact that the wrist pin load does not reverse direction, and thus does not facilitate squeeze film lubrication of the wrist pin bearing, special attention must be given to the wrist pin design. Applicable technology is found in the design of wrist pin bearings for two cycle Diesel engines. Several approaches are possible including the following:

1. Employment of needle type antifriction bearings.
2. Employment of a palm type bearing which provides for relatively low pressure loading over a large area on the outboard side of the piston pin.
3. Use of a helically grooved bushing in the connecting rod with lubricating oil force fed to the grooves, the wrist pin being fixed in the piston. Pitch of the grooves must be less than the amplitude of the rocking motion so that the motion drags oil from the grooves over the lands of the bearing.
4. Use of helically grooved bushings in the piston with the piston pin fixed to the connecting rod.

Consultation with the General Electric Diesel engine design group resulted in the selection of approach No. 4 above. Unit loadings of the order of 4000 psi were considered to be safe, since loadings in excess of 5000 psi have been successfully used in two cycle Diesel engines. Approach No. 4 permits lower loadings than approach No. 3. Except for this, the two are essentially identical.

2.1.3 Oil Free High Temperature Piston Ring Design

Friction, wear, and sealing effectiveness of the oil free piston rings represent the most important questions toward which the contract experimental program is directed. Basic design parameters of interest are the following:

1. Number of rings employed for 1000 psi sealing pressure
2. Width of rings

3. Use of pressure balancing to reduce total normal force and resulting friction and wear
4. Segmentation of rings and sealing between segments
5. Design of springs for backing of rings (non metallic)
6. Provision for positive assurance of sealing pressure behind the ring
7. Rubbing velocity.

An extensive survey was made in the literature and by contact with manufacturers and users of carbon and other types of high temperature oil free rings. No directly comparable application was identified. The most pertinent experience was that with high pressure oil free compressor piston rings. Recommendations for number of rings, and width of rings varied widely. Values selected do not represent the most conservative practice, but are the result of judgments which balance design for low friction and wear against the requirements of an acceptably simple and light weight design. In references 1, 2, and 3, some bases for analytical prediction of ring life were found, and these were applied (see below). The following general design features were applied:

1. For non metallic rings (graphite) two piece segmented construction is applied with a one piece spring ring backup. For metallic rings (coated) one piece construction is used.
2. Ring widths are approximately .3" for non metallic and .125" for coated metallic rings. Three non metallic compression rings are used for the crosshead design and two for the trunk design (because of smaller available space). For an alternate backup ring design, three coated metallic rings are used for both crosshead and trunk designs.
3. For non metallic rings, in which .060" to .080" radial wear is anticipated, holes are provided at the edge of the ring groove to assure positive access of sealing pressure behind the ring, thus preventing ring flutter.
4. Pressure balancing is provided with the nonmetallic rings to reduce average normal pressure, friction, and wear.

2.1.4 Piston Rod Sealing

In the crosshead piston design, the steam piston rod must seal the condenser steam pressure (20 psia) against atmospheric pressure, and also prevent the transport of oil into the steam. Metal temperature due to steam contact is reasonably low (approximately 300°F). Seal friction, however, may produce local high temperature. Some cooling is provided through the use of a hollow piston rod with oil splashing in the inside. Two alternate types of seals have been identified for such an application:

1. A Chevron cross section packing using filled teflon material.
2. A segmented carbon ring packing using a packing gland, backup seal rings, and garter springs.

The first is simpler and cheaper and will be used as a first trial design. The second, more conservative, is a backup design.

2.1.5 Oil Free Valve Stem Sealing

Two approaches were identified for valve stem sealing:

1. The use of multiple precision metallic rings for sealing between the valve stem and the valve stem guide housing.
2. The use of carbon ring packing in a packing gland surrounding the valve stem.

Since assurance of success seems high with (1), based on General Electric steam turbine valve sealing experience, and since it is simpler and less expensive than (2), this approach will be used on the test expanders.

2.1.6 General Reciprocating Engine Mechanical Design

Extensive consultations with engineers from Ford Motor Company and Waukesha Motor Company were conducted regarding reciprocating engine design practice in the areas of valve train and cam design, connecting rod and piston design, engine balancing, and bearing load capability analysis. It is to be noted that both test expanders are to be built around Waukesha test engine crankcases. These incorporate a crankshaft and main bearings of great robustness. Modification of these crankcases is required to produce a camshaft driven at full engine speed instead of half engine speed. This requires a replacement of the camshaft drive gear and the mounting of an idler gear to run between the crankshaft gear and the camshaft gear.

2.2 Expander Preliminary Design Studies

Prior to the design selection of detail features for the two single cylinder test expanders, studies were conducted in the following areas:

1. General cylinder configuration selection, cylinder sizing, and selection of speeds for testing to be conducted
2. Investigation of Ford Motor Company variable cut off mechanism
3. Cam shaft drive configuration
4. Valve and valve stem seal configuration
5. Wrist pin bearing configuration
6. Piston rod packing configuration
7. Piston ring configuration for crosshead and trunk piston designs.

These studies are demonstrated in Figures 2-1 through 2-7, and are briefly described below.

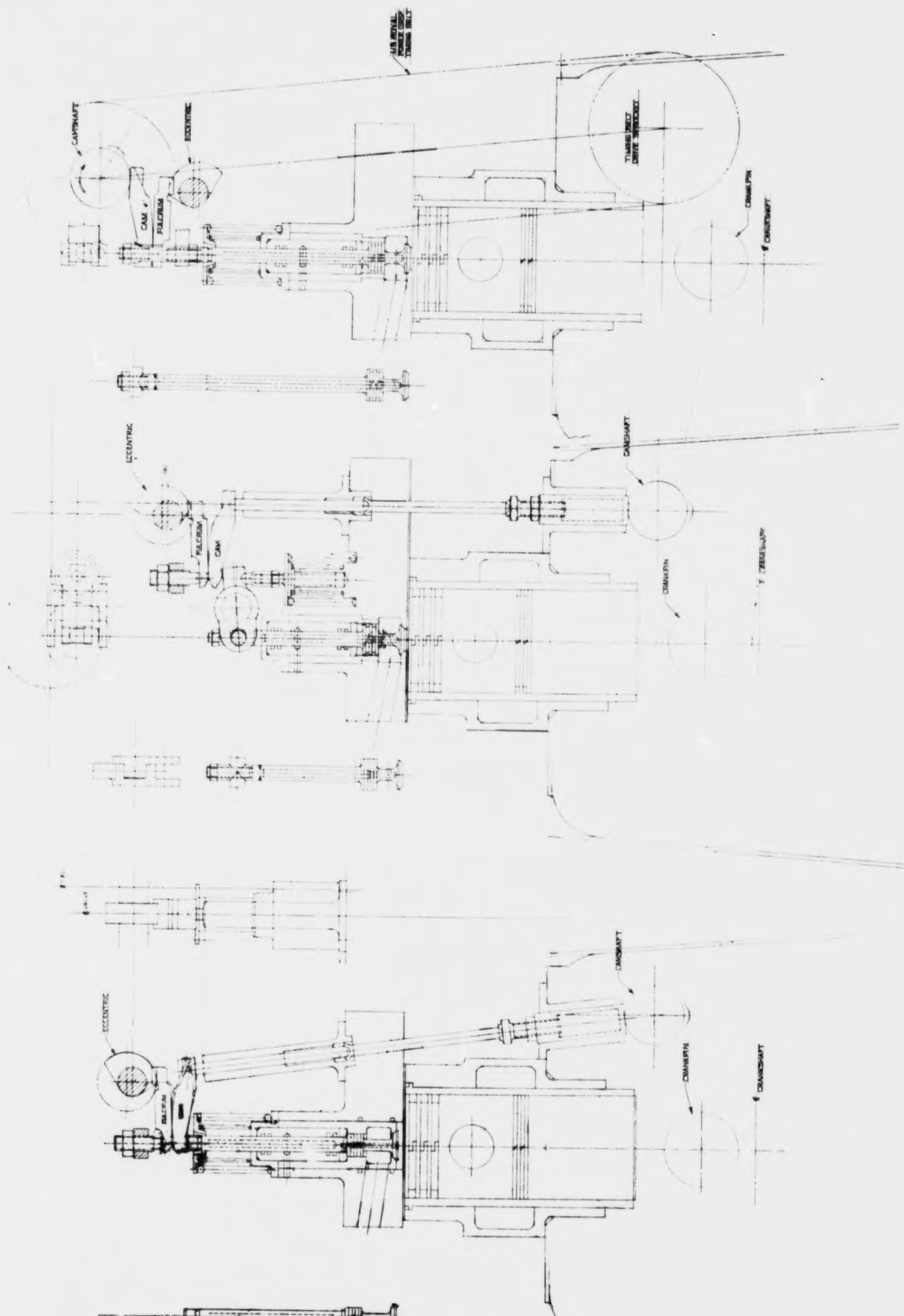
2.2.1 General Cylinder Configuration Selection

The principal alternatives with regard to the general cylinder configuration are that of a near zero clearance volume valve-in-head design with actuated inlet and exhaust valves both located in the cylinder head, and a uniflow engine configuration using exhaust ports opening at approximately 90% of stroke and an actuated inlet valve located either in the head or in a side inlet passage.

Advantages and disadvantages of these alternatives can be summarized as follows:

Zero Clearance Volume Design Advantages

1. Because of the exhaust valve can be kept open throughout the upward stroke of the piston, work or recompression can be virtually eliminated. This results in a higher MEP and a smaller cylinder displacement for a given output.


CAMSHAFT
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TIMING BELT DRIVE SPROCKET
CRANKPIN
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CRANKPIN
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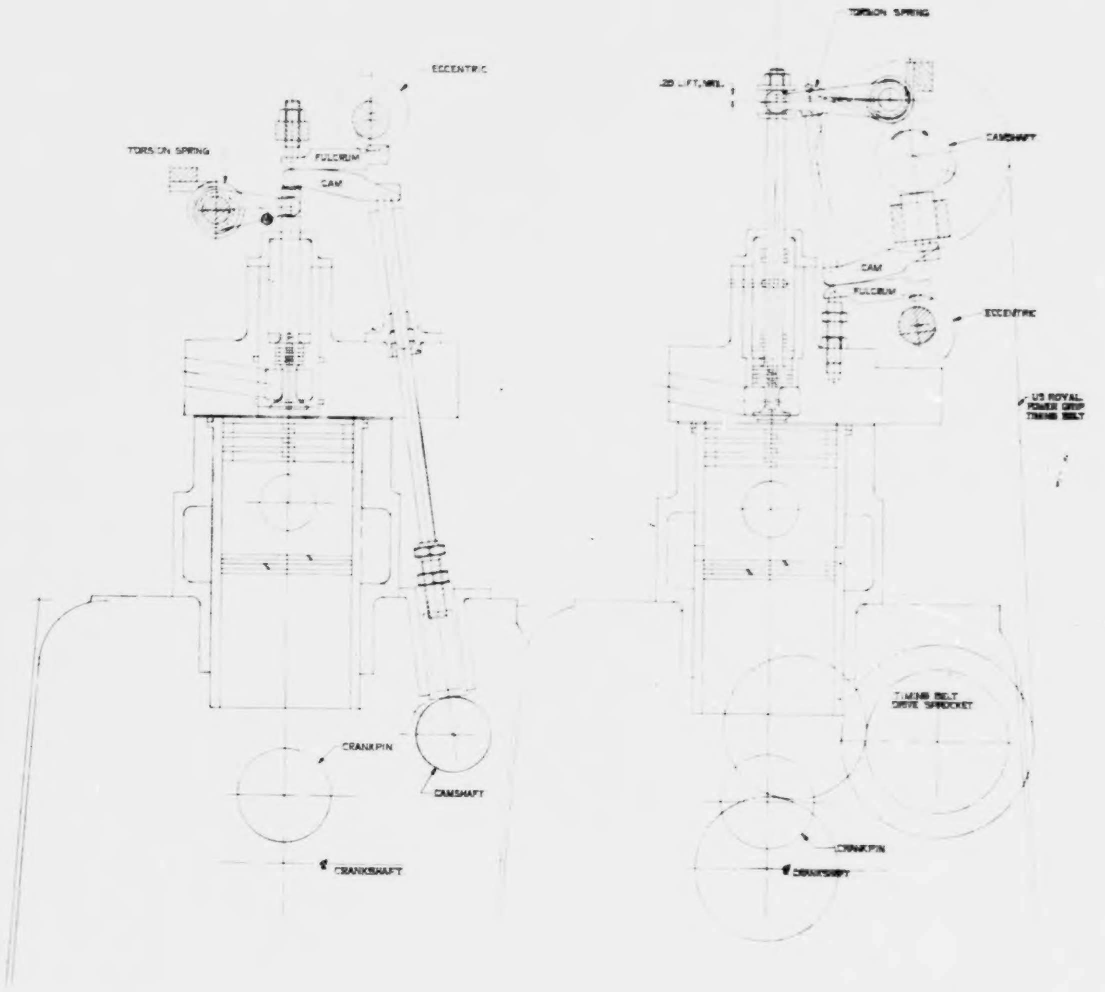
FULCRUM
CAMSHAFT
CRANKPIN
CRANKSHAFT

Figure 2-1. Valve - Variable Cut Off Mechanism.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CAM
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CRANKSHAFT
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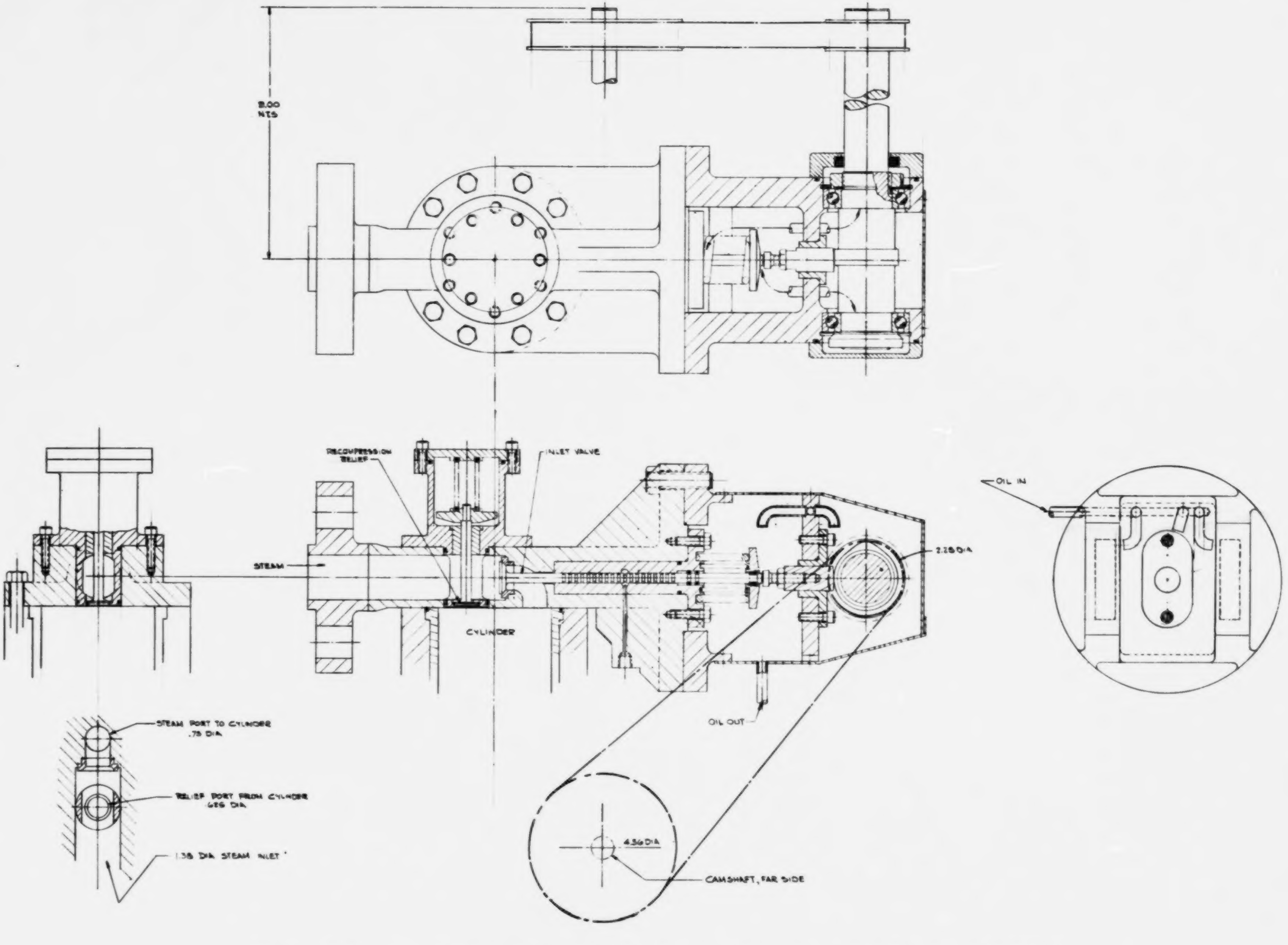
.20 LIFT, MIN.
CAMSHAFT
CAM
FULCRUM
ECCENTRIC
US ROYAL
POWER GRIP
TIMING BELT
TIMING BELT
DRIVE SPROCKET
CRANKPIN
CRANKSHAFT

Figure 2-2. Valve - Variable Cut Off Mechanism.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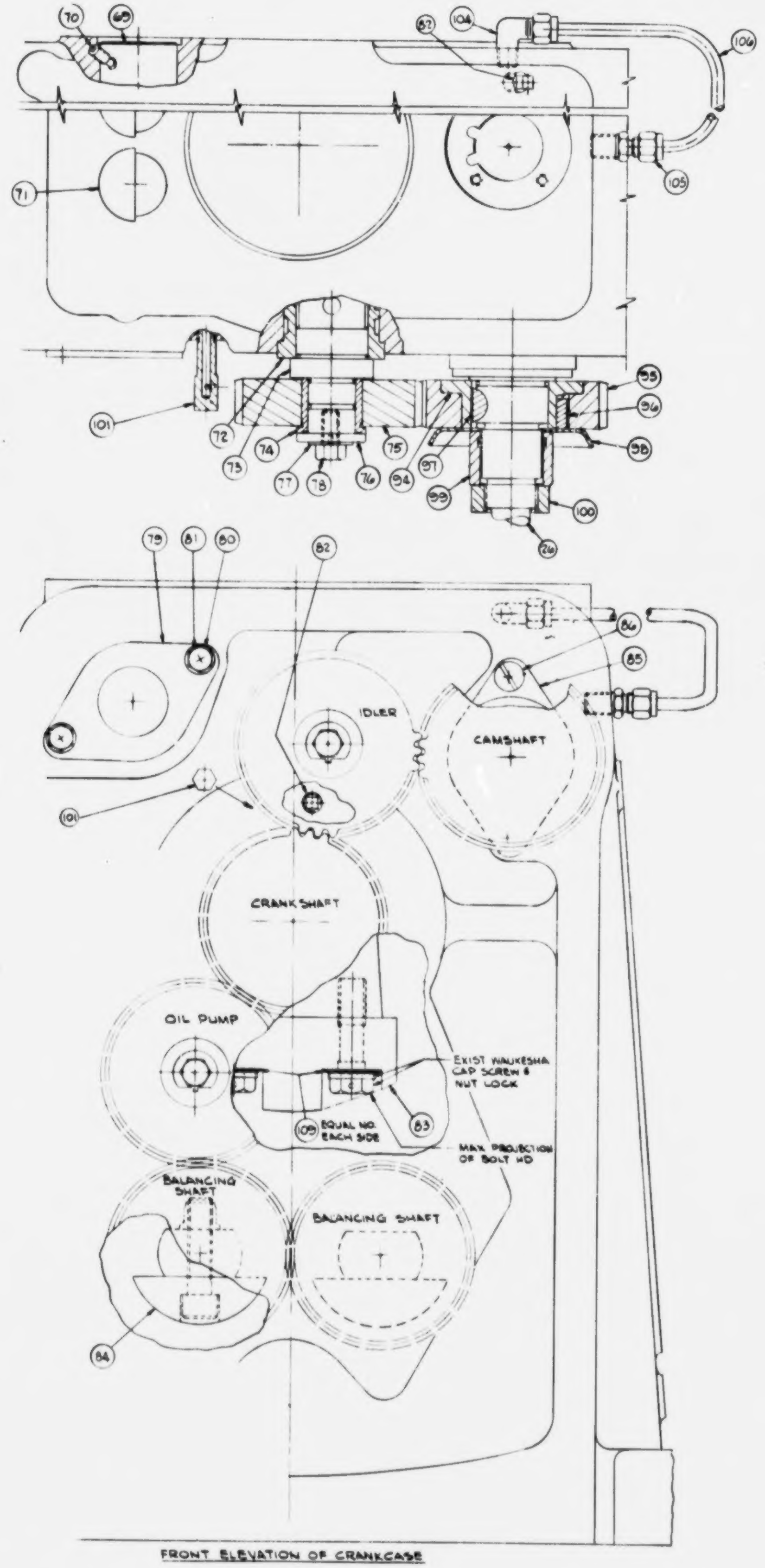
RECOMPRESSION RELIEF
INLET VALVE
STEAM
CYLINDER
2.25 DIA
OIL IN
OIL OUT
STEAM PORT TO CYLINDER .75 DIA
RELIEF PORT FROM CYLINDER .625 DIA
1.38 DIA STEAM INLET
.436 DIA
CAMSHAFT, FAR SIDE

Figure 2-3. Cam Shaft Drive, Head, Valve, and Cam Details (GE Dwg. 707E702).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100
26
79
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86
85
IDLER
CAMSHAFT
101
CRANK SHAFT
OIL PUMP
EXIST WAUKESHA CAP SCREW & NUT LOCK
109 EQUAL NO. EACH SIDE
83
MAX PROJECTION OF BOLT HD
BALANCING SHAFT
BALANCING SHAFT
84
FRONT ELEVATION OF CRANKCASE

Figure 2-4. Cam Shaft Drive, Steam Expander Trunk Type Piston (GE Dwg. 707E719, Sh. 2)

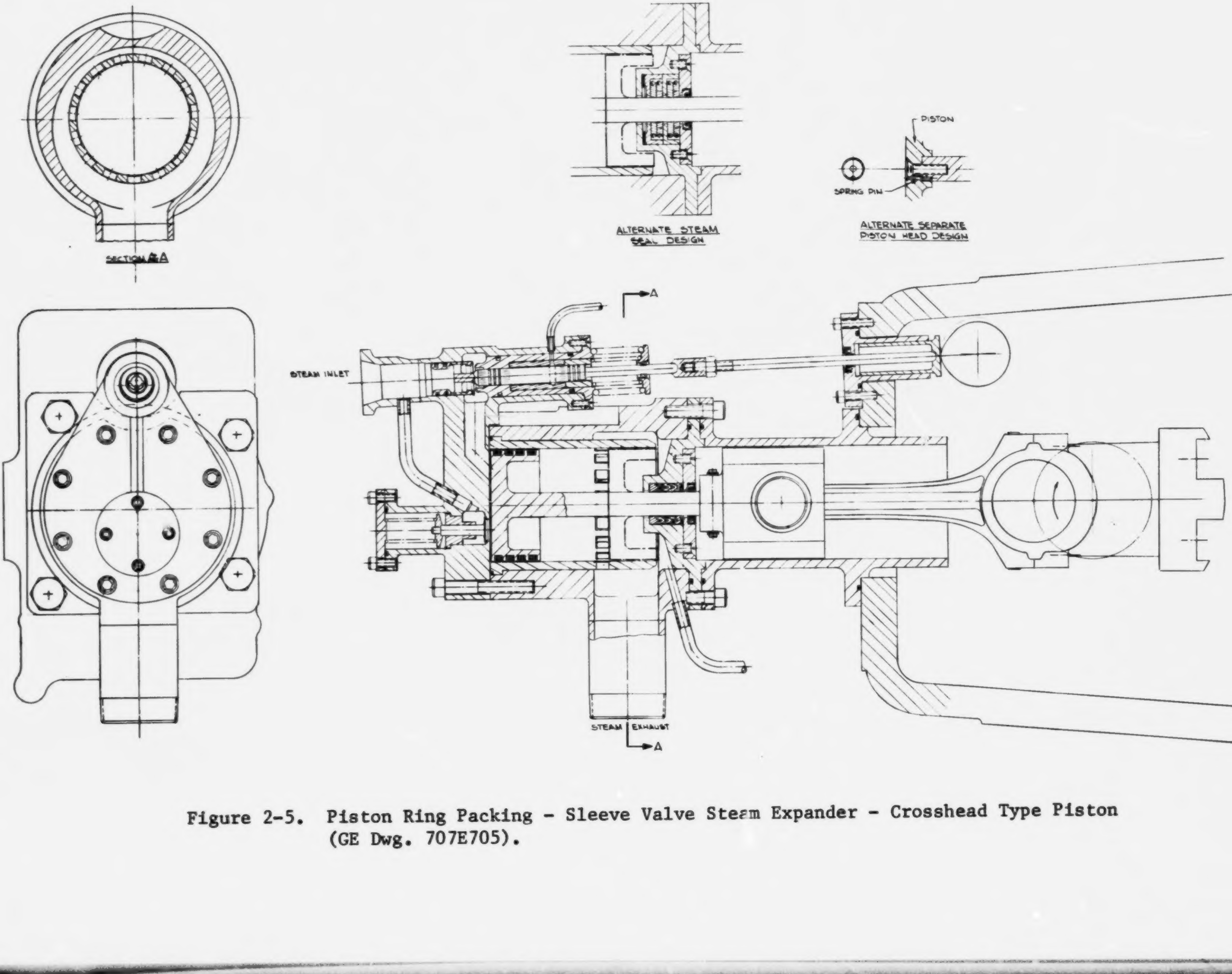

SECTION A-A
ALTERNATE STEAM SEAL DESIGN
PISTON
SPRING PIN
ALTERNATE SEPARATE PISTON HEAD DESIGN
A
STEAM INLET
STEAM EXHAUST
A

Figure 2-5. Piston Ring Packing - Sleeve Valve Steam Expander - Crosshead Type Piston (GE Dwg. 707E705).

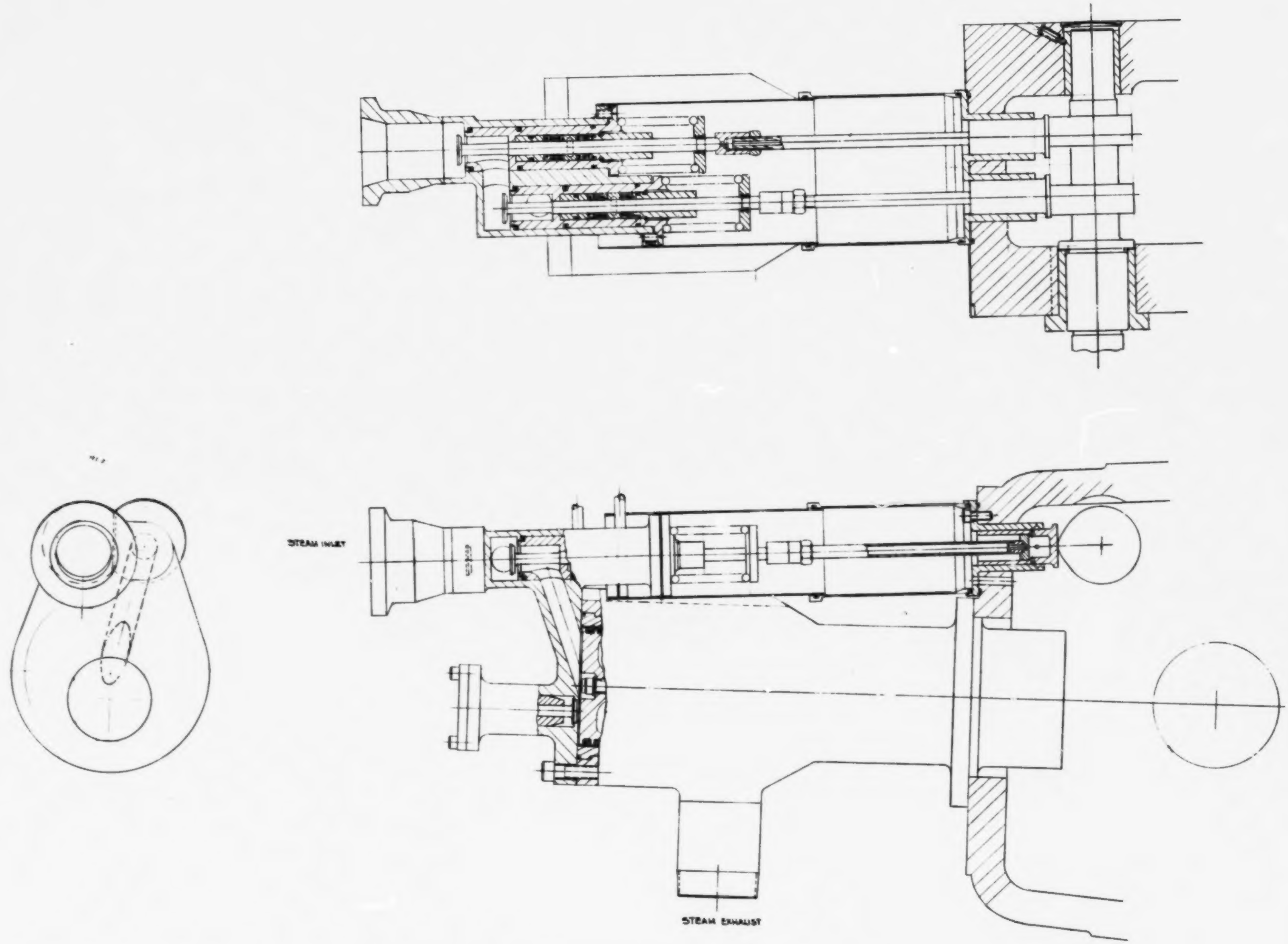

STEAM INLET
STEAM EXHAUST

Figure 2-6. Valve Sleeve Packing - Cutoff Valve Steam Expander - Crosshead Type Piston (GE Dwg. 707E710).

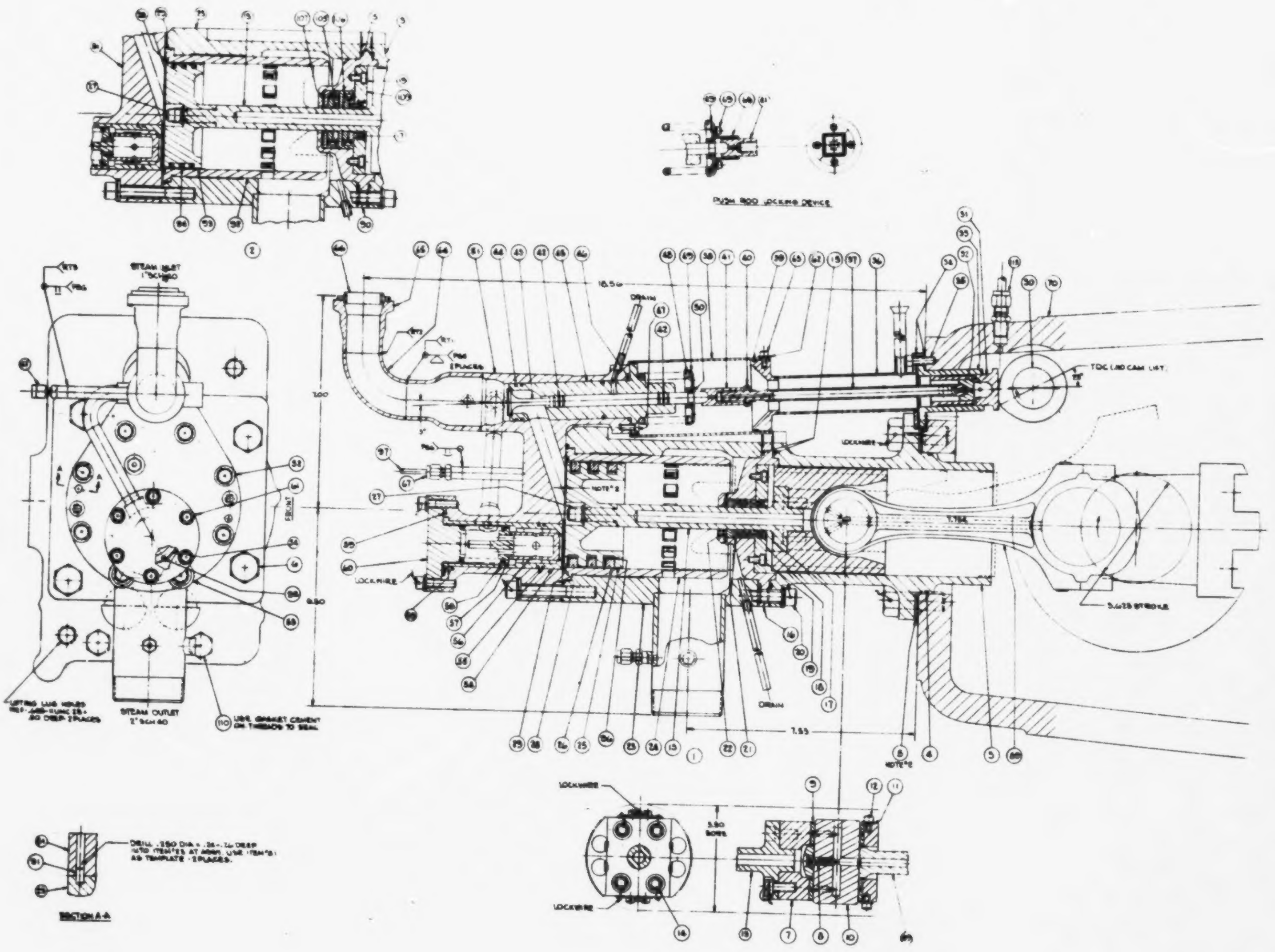

PUSH ROD LOCKING DEVICE
STEAM INLET
STEAM OUTLET 2" SCH 80
DRAIN
DRAIN
LOCKWIRE
LOCKWIRE
LOCKWIRE
LOCKWIRE
NOTE 2
NOTE 2
FRONT
18.50
7.55
3.80 BORE
5.625 STROKE
TDC LAD CAM LIFT
LUBE GASKET CEMENT ON THREADS TO SEAL
SECTION A-A

Figure 2-7. Steam Expander Assembly - Crosshead Type Piston (GE Dwg. 707E713, Sh. 1).

Reproduced from best available copy.

2. Because expansion can be continued to the end of the downward stroke of the piston, the effective expansion ratio can be increased. This is a factor favorable to high efficiency.

Disadvantages

1. Cooling of the upper part of the cylinder occurs during steam exhaust. This results in "condensation loss" during the subsequent expansion.
2. High thermal gradients occur in the cylinder head due to the proximity of the inlet and exhaust steam. This is unfavorable from the standpoint of mechanical design reliability.
3. The need for an actuated exhaust valve in addition to an actuated inlet valve results in undesirable complexity.

Uniflow Design Advantages

1. The elimination of cylinder cooling and related losses associated with reverse flow exhaust. With proper design control of the clearance volume, efficiency can be kept at least equal to that of the zero clearance volume design.
2. A maximum of design simplicity.

Disadvantages

1. Larger cylinder displacement for a given output and efficiency.
2. The need for a spring loaded compression relief valve to prevent recompression pressure from exceeding the inlet pressure and damaging the valve train when the inlet pressure is low and/or the condenser pressure is high.

Largely from considerations of mechanical reliability and simplicity, it was decided to choose the uniflow approach.

For a uniflow engine, thermodynamic sizing calculations indicated that for 170 IHP at 2500 RPM an expander indicated adiabatic efficiency of approximately 80% could be achieved at an inlet pressure and temperature of 1000 psi, and 1000°F, and an exhaust pressure of 20 psi. This requires a clearance volume of about 5% and exhaust porting opening at

90% of stroke. Effective cut off is approximately 15%. Bore is approximately 3-1/2" and stroke is approximately 3-1/2". This result, combined with investigation of available sizes of test engine crankcases led to the selection of a cylinder bore of 3-1/2" and a stroke of 3-5/8". This stroke was set by the available (Waukesha) crankshaft dimensions.

Engine breathing analysis, described below, indicated that a single valve train could be used to provide steam cut off to the level of the steam supply limit (400 lb/hr) at speeds up to 2500 RPM. A change in valve lead angle from 10° to 20°, however, is necessary to run at 2500 RPM within the steam supply limit. It was agreed that tests should be carried out for 150 hours at 2000 RPM, and 150 hours at 2500 RPM.

2.2.2 Investigation of Ford Motor Company Variable Cut Off Mechanism

A survey of alternate variable cut off mechanisms led to the conclusion that it might be feasible and advantageous to make use of the Ford Motor Company variable cut off mechanism on the test expanders. It was felt that this could be done without incorporating the camshaft phasing gear, leaving this step for future effort. Accordingly, a design study of alternate forms of this mechanism was made. This work was carried out at the Ford Plant. Results are shown in Figures 2-1 and 2-2. The essential feature of the mechanism is an overhead rocker arm valve actuating linkage in which the position of the effective pivot about which the rocker arm actuating the valve turns can be varied by the adjustment of an eccentric bearing upon a fulcrum link. This changes the amplitude and event angle of the valve motion. As shown by Figures 2-1 and 2-2, the mechanism can be arranged to permit either inward or outward opening of the valve. However, for inward valve operation it is essential that a balanced valve be used to prevent steam inlet pressure from opening the valve.

A further problem is the need for oil lubrication of the contact interface between the valve actuating rocker and the adjustable fulcrum piece.

Perhaps the greatest problem in the Ford mechanism for the steam car application is the increase in overall expander height which it involves. This is a problem common to all valve in head designs.

2.2.3 Cam Shaft Drive

In conjunction with the use of purchased Waukesha crankcases for the test expanders, it becomes necessary to select a camshaft drive. The principal alternatives considered were the use of an overhead or side mounted camshaft and the use of a crankcase mounted camshaft. The former would be belt driven from a shieve mounted on the Waukesha camshaft (which turns at 1/2 crankshaft speed), while the latter would be driven from the crankshaft through a modified gear train designed for a crankshaft to camshaft speed ratio of unity. These alternate arrangements are illustrated in Figures 2-3 and 2-4. For the crankcase mounted cam configuration, which is obviously much neater, it is necessary to replace the camshaft drive gear with a smaller gear and to mount an idler gear between the crankshaft and camshaft gears.

2.2.4 Valve and Valve Stem Seal

Considerable attention was given to the selection of the valve configuration, and to the design of the oil free valve stem seal. Valve types investigated included the double-beat and balance piston types of balanced poppet valve, a ring sealed sleeve type valve, and the simple unbalanced poppet valve. The advantage of the balanced type valves is that they reduce the steam pressure loading on the valve train, and, specifically upon the cam - cam follower interface. The disadvantages of these valves are complexity and leakage. After detail sizing of the simple unbalanced poppet valve, and calculation of the cam-tappet Hertz stress it was decided that this valve configuration is quite acceptable.

Alternate types of valve stem seal are illustrated in Figures 2-5 and 2-6. (Figure 2-5 also illustrates the sleeve type partially balanced valve). These are the metallic piston ring seal and the graphite packing ring seal. The former enjoys a substantial advantage in elegance and simplicity. Upon the identification of satisfactory materials, (see below) this approach was accepted. It will be noted that the valve stem seal is divided into two parts with a drain to the condenser between them. This approach minimizes the pressure differential across the seal to the atmosphere, thus minimizing leakage from the steam system.

2.2.5 Wrist Pin Bearing

As mentioned above the alternate types of wrist pin bearing considered included the needle type antifriction bearing and the helical groove bearing. Both of these types were sized and laid out. Largely on the basis of the experience and recommendation of the General Electric Diesel engine design group the helical groove bearing was selected.

2.2.6 Piston Rod Packing

The piston rod packing alternatives (crosshead design) have been discussed above. These include the teflon filled Chevron packing and the segmented carbon ring packing. These designs are illustrated in Figure 2-5.

2.2.7 Piston Rings

Design considerations relating to the compression rings have been discussed above. Two alternate designs, a graphite ring, and a coated metallic ring have been selected, the second design to be used as a backup in the event of a failure with the first. These designs are shown on Figures 2-7 and 2-8.

In the trunk piston expander, shown in Figure 2-8, in addition to compression rings, a graphite rider or side thrust carrying ring, and, on the bottom of the piston skirt two oil scraper rings are provided. The rider ring is made in two parts. A side thrust bearing section, which will be oil lubricated, is provided at the extreme bottom of the piston skirt. Thus, side thrust bearing areas are provided both above and below the wrist pin. Above the oil scraper rings on the trunk piston is a metallic compression ring to seal condenser pressure.

2.3 Selected Expander Configurations

Overall assembly drawings of the selected crosshead and trunk piston expander designs are shown in Figures 2-7 and 2-8. Salient features of these designs may be summarized as follows:

1. Uniflow design with one unbalanced cam actuated poppet valve, and a spring loaded recompression relief valve.

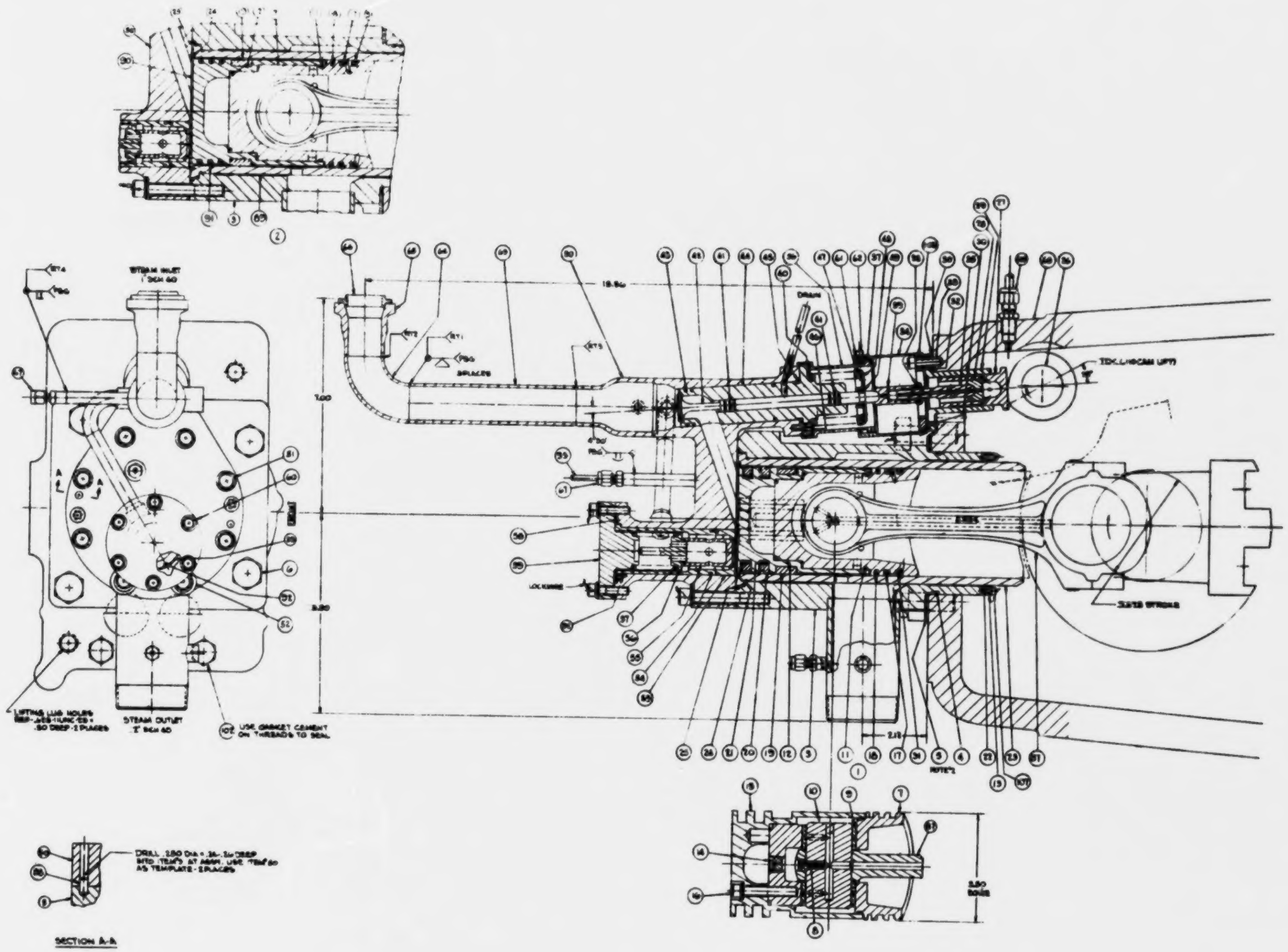

STEAM INLET
1" SCH 60
STEAM OUTLET
2" SCH 60
LIFTING LUG HOLES
USE GASKET CEMENT ON THREADS TO SEAL
7.00
DRAIN
LOCKWIRE
SECTION A-A
2
1

Figure 2-8. Steam Expander Assembly - Trunk Type Piston (GE Dwg. 707E719, Sh. 1).

2. Straight line valve train actuated from a crankcase mounted camshaft. Mushroom type cam with offset (for tappet rotation) flat faced cam follower. Valve lead is to be increased for high speed testing by changing camshaft gear mesh by 1 tooth. This reduces the effective cut off.
3. 5% clearance volume, most of it in the side steam admission passage.
4. Exhaust porting in the cylinder liner opening between 135° and 225° crank angle measured from T.D.C.
5. Helical groove wrist pin bearing bushings mounted in the piston.
6. Alternate non metallic or coated metallic compression piston rings.
7. Non metallic rider ring on trunk piston.
8. Metallic piston ring sealed valve sleeve.
9. Alternate teflon filled and carbon packing ring piston rod packing (crosshead design).

2.4 Expander Performance Analysis

2.4.1 Expander Breathing Analysis

In order to select the expander inlet valve design so as to keep the steam flow at a value in the range of 80% to 90% of the test facility steam supply (400 lbs per hour), and also to calculate the indicated expander horsepower and efficiency, a computer program was written and applied to the breathing process of the uniflow engine. The calculation procedure was based on an energy balance for the processes occurring during intervals of time corresponding to 0.25 degree of crankshaft rotation. Flow through the inlet valve was based upon constant entropy and total entalphy between inlet pipe stagnation conditions and static conditions at the nozzle throat. The assumed value of inlet nozzle flow coefficient was 0.68. Cylinder pressure was assumed to be equal to the nozzle throat static pressure. Cylinder wall heat transfer and piston ring blow-by were neglected.

Engine indicator diagrams for the two selected expander test conditions are shown in Figures 2-9 and 2-10. Thermodynamic data relating to these diagrams is given in Table 2-1.

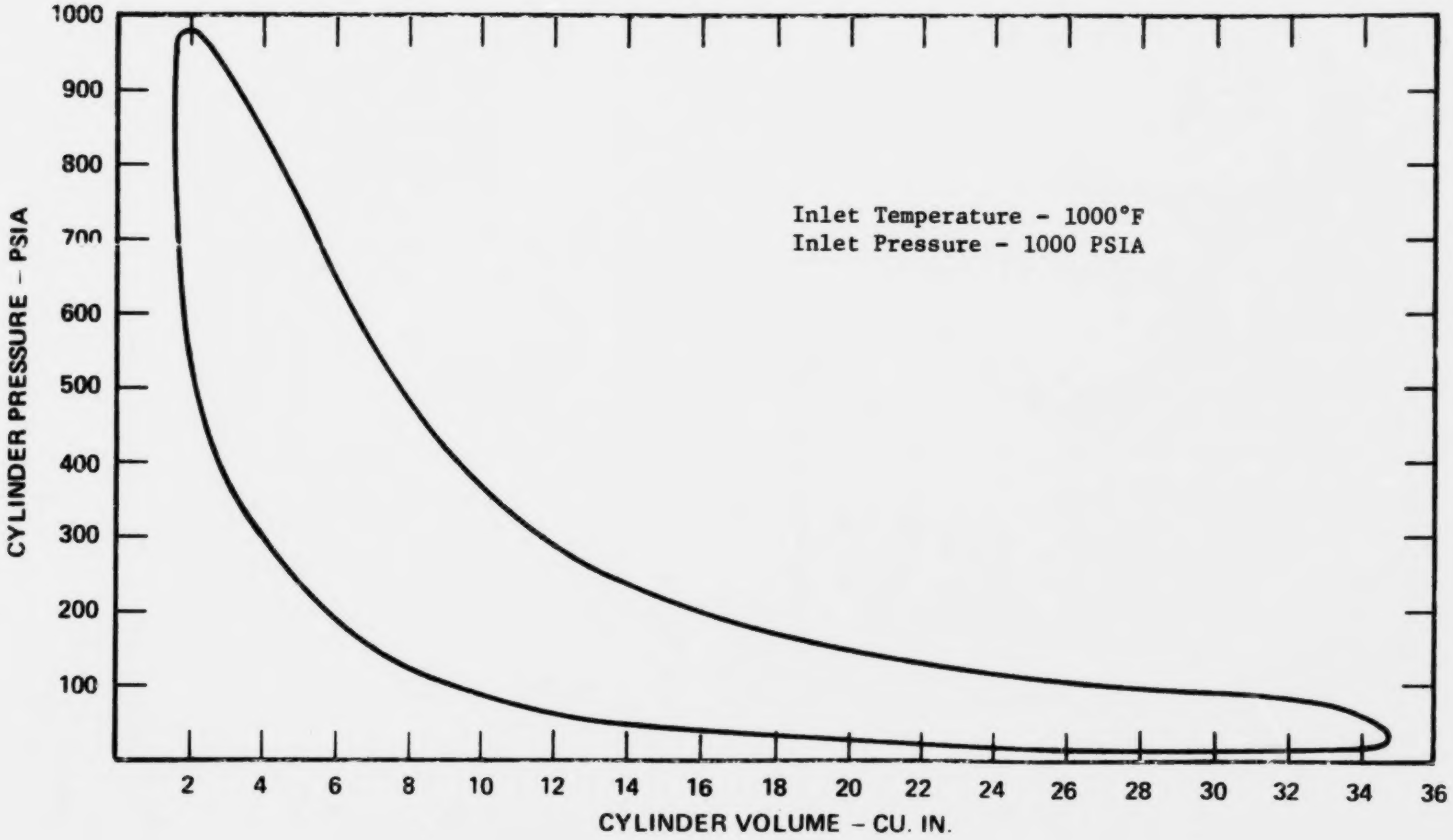


Figure 2-9. Expander Indicator Diagram, 2500 RPM, 20° Valve Lead.

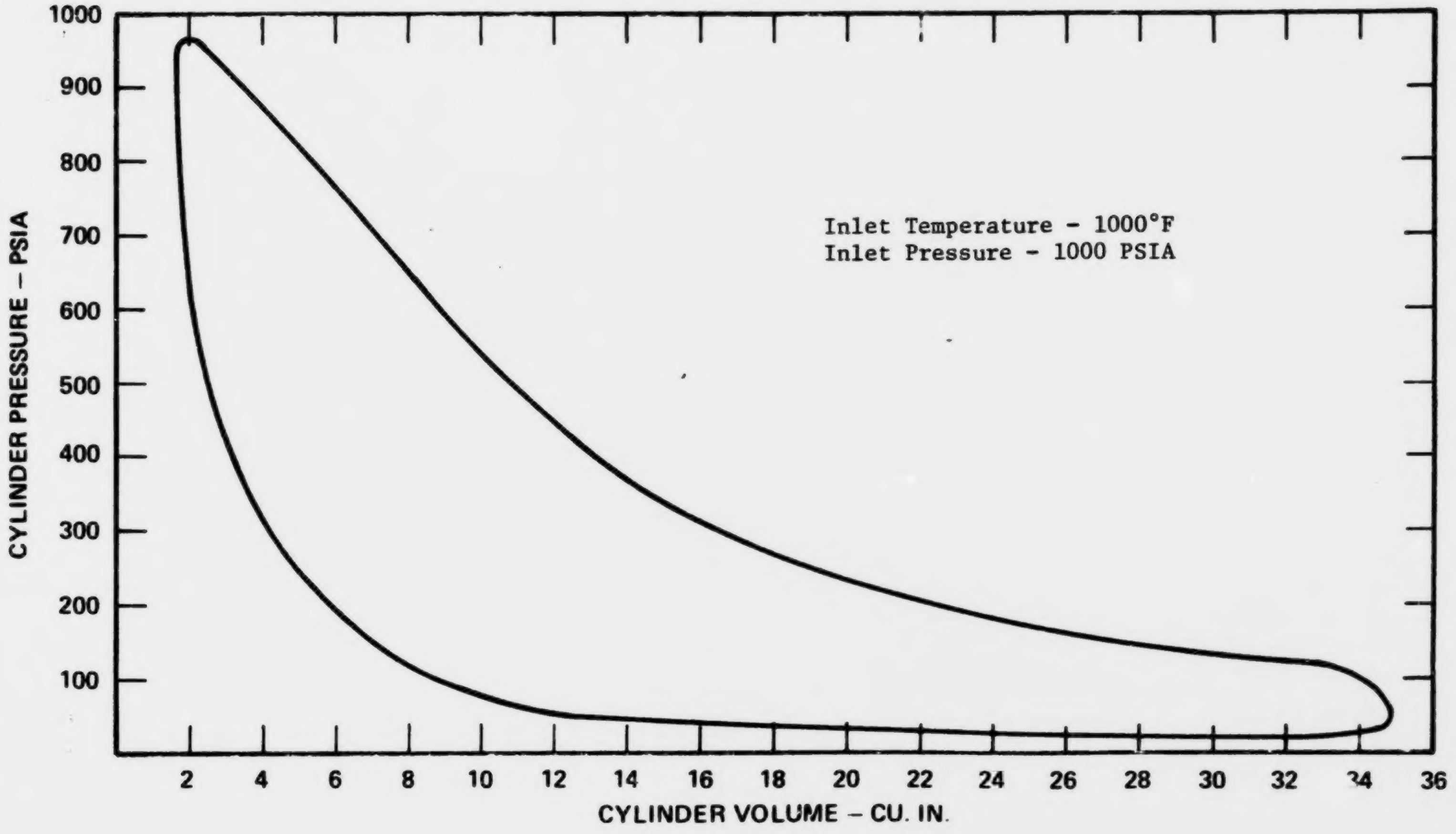


Figure 2-10. Expander Indicator Diagram, 2000 RPM, 10° Valve Lead.

Table 2-1

Breathing Analysis Summary

Indicator Diagram	Fig. 2-9	Fig. 2-10
Valve Event	70°	70°
Valve Lead	20°	10°
Valve Throat Diameter	0.75 in.	0.75 in.
Maximum Lift	0.10 in.	0.10 in.
RPM	2500	2000
Steam Flow Rate	332 lb/hr	340 lb/hr
Indicated Horsepower	41.7	44.5
Indicated Expander Efficiency	79%	82.5%

2.4.2 Steady-State Temperature Distribution

A broad heat transfer analysis was conducted to determine an order of magnitude temperature map for the two single-cylinder steam expanders employing the existing THT-D digital computer program. This program is capable of analyzing general three-dimensional heat transfer systems employing a finite difference method with appropriate specified boundary conditions. A variety of modes of heat exchange may be analyzed including heat transfer by conduction, convection, and radiation with or without internal heat generation or surface heat flux.

Special attention was given to piston ring, piston and cylinder heat generation and temperature as a result of steam temperature, frictional heat generation, and to the boundary conditions which are expected to exist during expander operation.

Complex geometrical areas in the expanders were replaced with simple geometry to facilitate the node assignment for the computer calculations. This approximation was made without changing the basic design. After dividing the geometry into a series of numbered nodes, measurements were made of full scale drawings and the necessary areas and volumes for each node were calculated and served as input to the THT-D computer program. Assumptions and boundary conditions for the computer analysis are listed below:

1. Due to symmetry, it was assumed that no heat flowed across the centerline or through the circumferential faces.
2. Thermal insulation on the outside of the expander was simulated with an appropriate convection coefficient.
3. The ambient air temperature was taken as 80°F, and the film coefficient for the surface exposed to air was estimated by appropriate equation for natural convection over vertical cylindrical surface.
4. Steam temperatures were taken as 1000°F in the intake valve region and 540°F in the piston/cylinder region. The exhaust steam temperature was taken as 250°F.
5. The convective heat transfer coefficient for steam was estimated from conventional equations such as the well known Dittus-Boelter equation for flowing fluid. In the region

enclosed by the cylinder and the top of the piston where some degree of steam condensation might occur, the film coefficient was estimated by equations appropriate for film condensation over a flat plate.

6. The cooling oil temperature was taken as 250°F, and the film coefficients were estimated from equations appropriate for fluid flowing through tubes or over vertical surfaces.
7. All contact coefficients were assumed to be 20,000 $\mathrm{Btu/hr\text{-}ft^2\text{-}°F}$.

There are five significant places where metal parts sliding over each other generate heat through friction. These five places are:

1. Graphite ring with cylinder lining (graphite-steel)
2. Crosshead with cylinder (aluminum-steel)
3. Teflon packing with piston rod (teflon-steel)
4. Rider ring with cylinder lining (graphite-steel)
5. Intake valve ring with valve guide (steel-steel).

The heat generated by friction is calculated from the following equation:

$$Q_f = \left(\frac{f}{J}\right) \frac{1}{2\pi} \int_0^{2\pi} (F.u)\, d\theta$$

where f = sliding coefficient of friction

J = 778 ft-lb/Btu

F = Normal force acting on the sliding surface, a function of θ

u = Sliding velocity, a function of θ

θ = Crank angle

In general the frictional heat generated by a variable normal force and a variable sliding velocity is a function of crank angle. For steady state calculation, however, the above equation can be used to calculate the frictional heat, and can be considered as the mean effective frictional heat generated by two sliding parts.

Typical values of sliding coefficient of friction, f, in the above equation as cited from the latest Mechanical Engineering Handbook - 7th edition, are listed as follows:

Sliding Parts Material	Sliding Coefficient of Friction	
	Dry	Grease
steel on steel	0.42	0.03-0.1
steel on graphite	0.21	0.09
teflon on steel	0.04	----

The frictional heat generated by various sliding parts for an expander speed of 2500 RPM are calculated and summarized as follows:

	Sliding Parts	Total Heat Generated by Friction, Btu/hr
1)	Graphite Ring - Cylinder (f = 0.1 and F = 3 lbf/in of circumference)	305
2)	Crosshead - Cylinder (f = 0.07)	1980
3)	Teflon packing - piston rod (f = 0.04 and F = 3 lbf/in of circumference)	26.2
4)	Rider Ring - Cylinder (f = 0.1)	2840
5)	Intake valve - valve guide (f = 0.4 and F = 3 lbf/in of circumference)	170

Regarding how the generated total frictional heat was distributed to the sliding parts, an assumption was made to split this evenly between the stationary part and the moving part. For example, for the crosshead-cylinder case above, 990 Btu/hr goes into the crosshead and 990 Btu/hr goes into the cylinder body.

Steady state temperature distributions were obtained from THT-D program computations, and are shown in Figure 2-11 (Drawing 707E726) for the crosshead expander and are shown in Figure 2-12 (Drawing 707E727) for the trunk piston expander. The temperature levels shown are order of

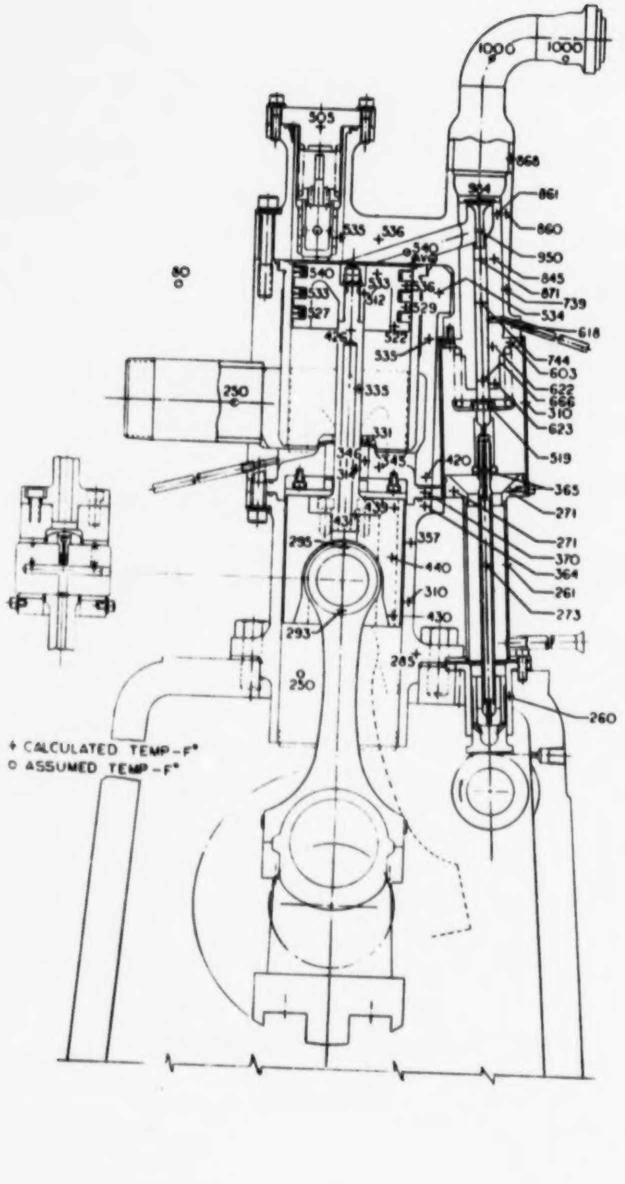
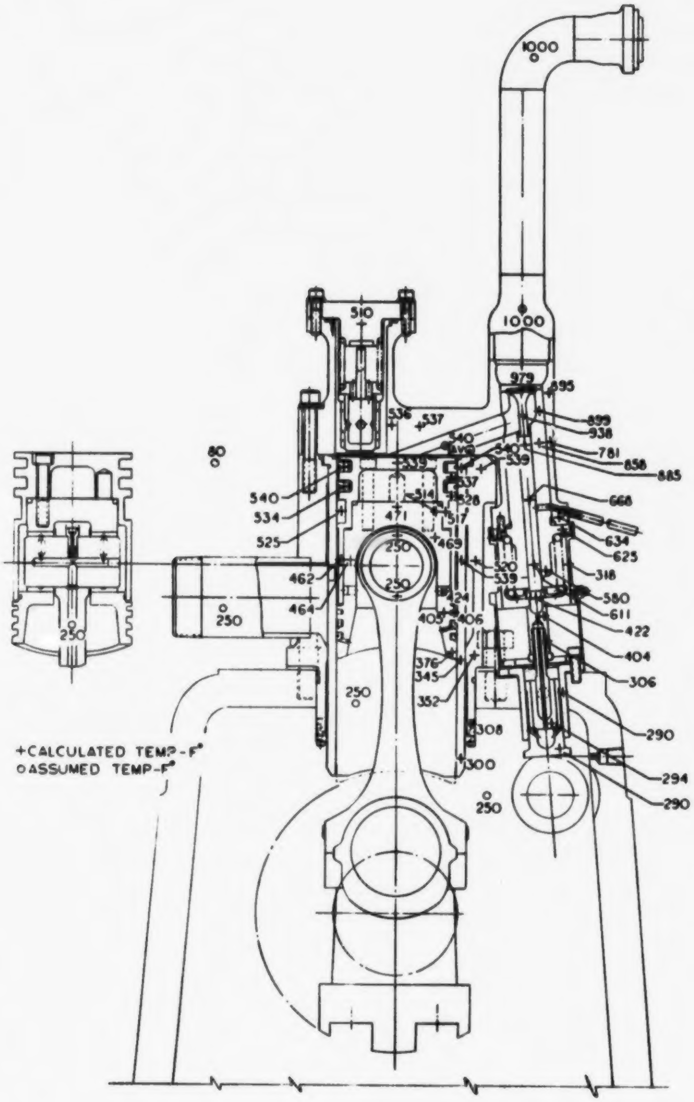

1000
1000
505
868
904
861
860
950
845
871
739
534
618
744
603
622
666
310
623
519
365
271
271
370
364
261
273
260
335
336
540
540
533
527
533
512
536
529
522
426
535
335
250
80
331
346
345
314
420
439
431
295
357
440
310
430
293
285
250
+ CALCULATED TEMP-F°
○ ASSUMED TEMP-F°

Figure 2-11. Temperature Profile for Crosshead Type Piston (GE Dwg. 707E726).



1000
1000
510
979
895
536
537
899
938
540 AVG
540
339
781
858
885
339
537
514
528
666
80
540
534
525
471
517
469
634
625
250
318
320
339
462
464
250
250
250
580
611
424
405
406
422
404
376
345
306
352
250
290
308
+CALCULATED TEMP-F°
○ASSUMED TEMP-F°
300
294
290
250

Figure 2-12. Temperature Profile for Trunk Type Piston (GE Dwg. 707E727).

magnitude values since a rigorous thermal analysis was not considered essential.

2.4.3 Stress Analysis

2.4.3.1 Crosshead Piston Expander

The piston position, velocity, and accelration; cylinder swept volume; and inertia force are shown on Figure 2-13 as functions of crank angle for a rotative speed of 1000 rpm and the 6.146 lb. reciprocating weight which acts on the wrist pin.

The pressure forces on the piston were calculated using a cycle with 15% cut off (∼ 40° ATDC) and isentropic expansion according to

$$PV^{1.28} = 1000\ (2.747)^{1.28} = \text{constant}$$

from 40° to exhaust port opening at $\theta = 135°$. The pressure remains at the condenser pressure of 20 psia until exhaust port closing at 225°. Isentropic compression occurs from 225° to 360° according to

$$PV^{1.28} = 20\ (32.535)^{1.28} = \text{constant.}$$

This model pressure cycle, shown in Figure 2-14 results in a mean pressure of 290 psi and a mean temperature of 575°F during the cycle.

The vertical pressure and inertia forces on the wrist pin are shown on Figure 2-15 as functions of crank angle. By combining the two forces, the total vertical force on the wrist pin under conditions of zero inlet pressure (motoring) or for 1000 psia inlet pressure for rotative speeds of 2000 and 2500 RPM can be determined.

The pressure and inertia forces at 2500 RPM on the wrist pin are combined and converted into a side force as a function of the crank angle in Figure 2-16. This data is useful in determining the total wrist pin load, with Figure 2-15. It also represents the side load on the crosshead piston.

The vectorally combined vertical and side forces on the wrist pin are shown on Figure 2-17 for a rotative speed of 2500 RPM. The forces are converted to projected pressures using the projected bearing area of 2.25 in^2.

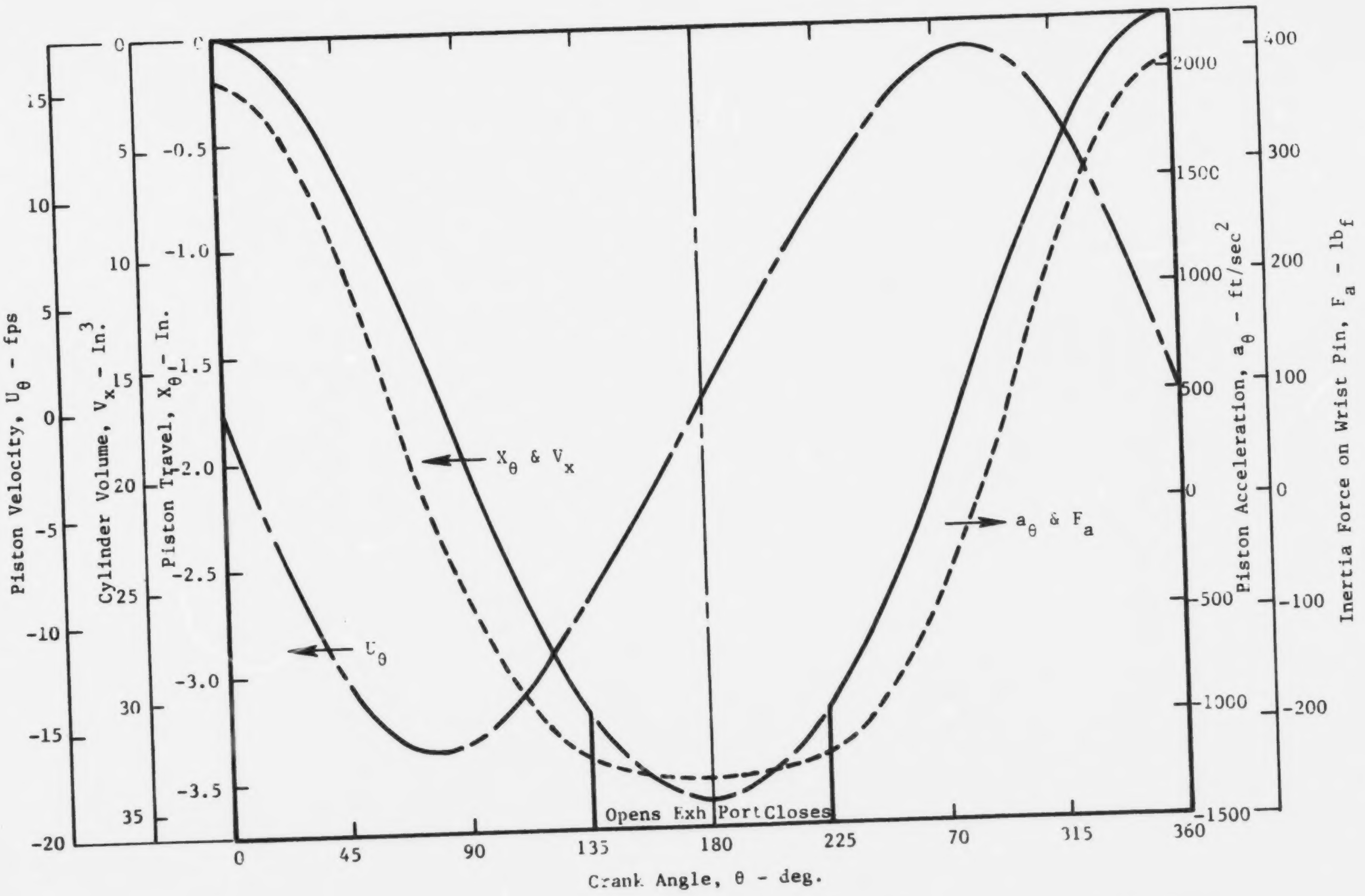


Figure 2-13. Piston Position, Velocity, and Acceleration; Cylinder Swept Volume; and Inertia Force for 1000 RPM and 6.146 lb. (Reciprocating Weight Acting on Wrist Pin) - Crosshead Expander.

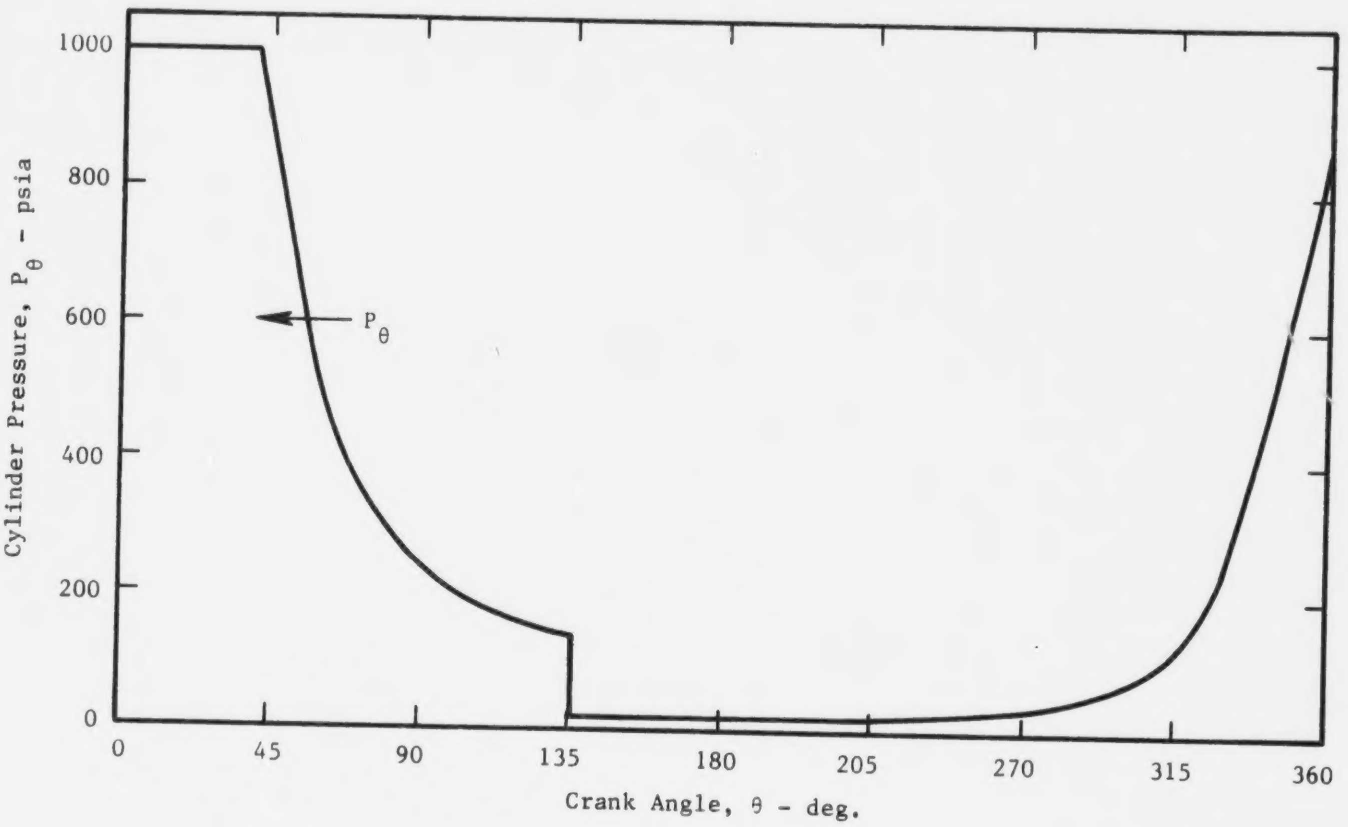


Figure 2-14. Model Cylinder Pressure Cycle - Crosshead Expander.

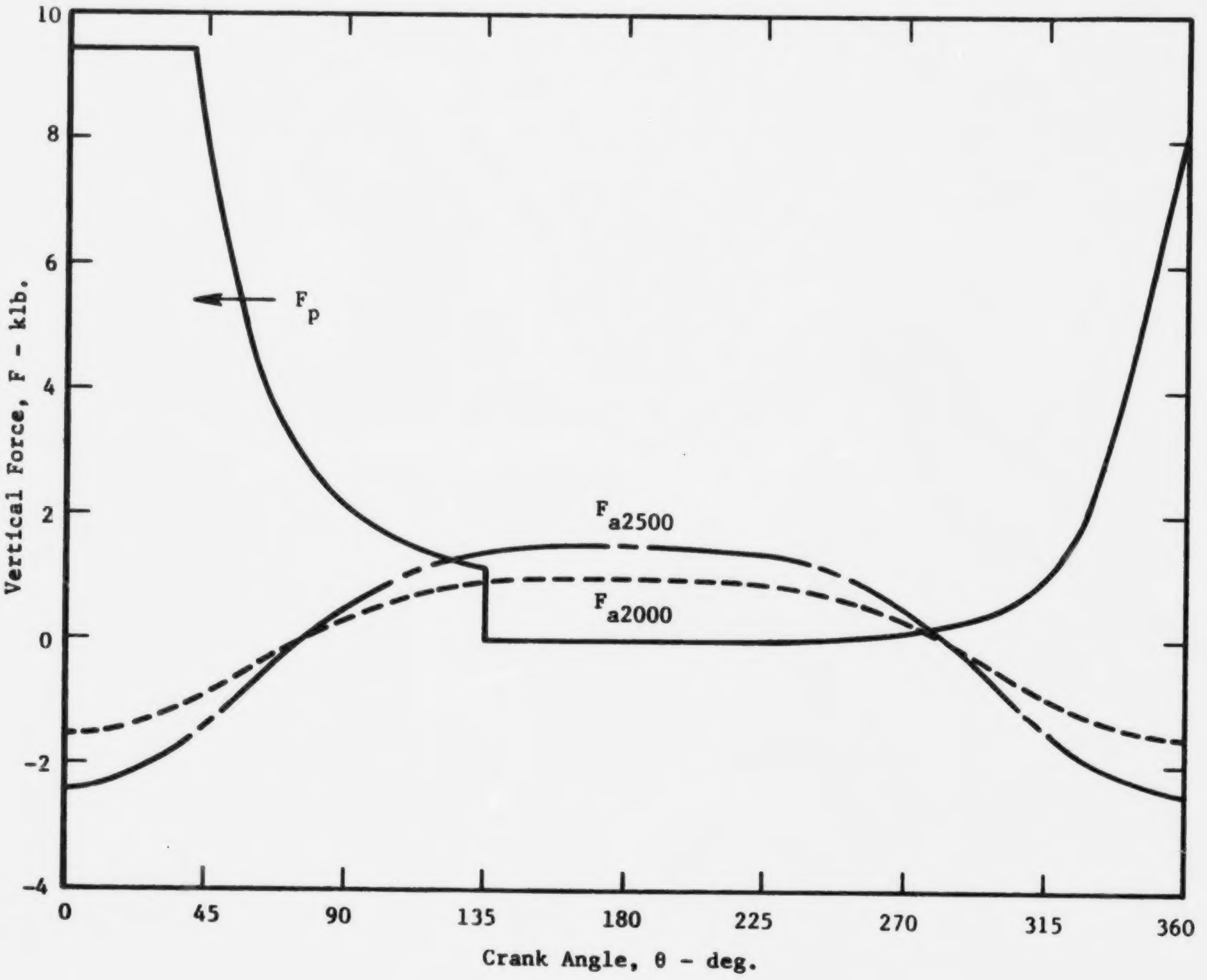


Figure 2-15. Vertical Wrist Pin Forces - Crosshead Expander. F_p = Pressure Force. F_a = Inertia Forces at 2000 and 2500 RPM.

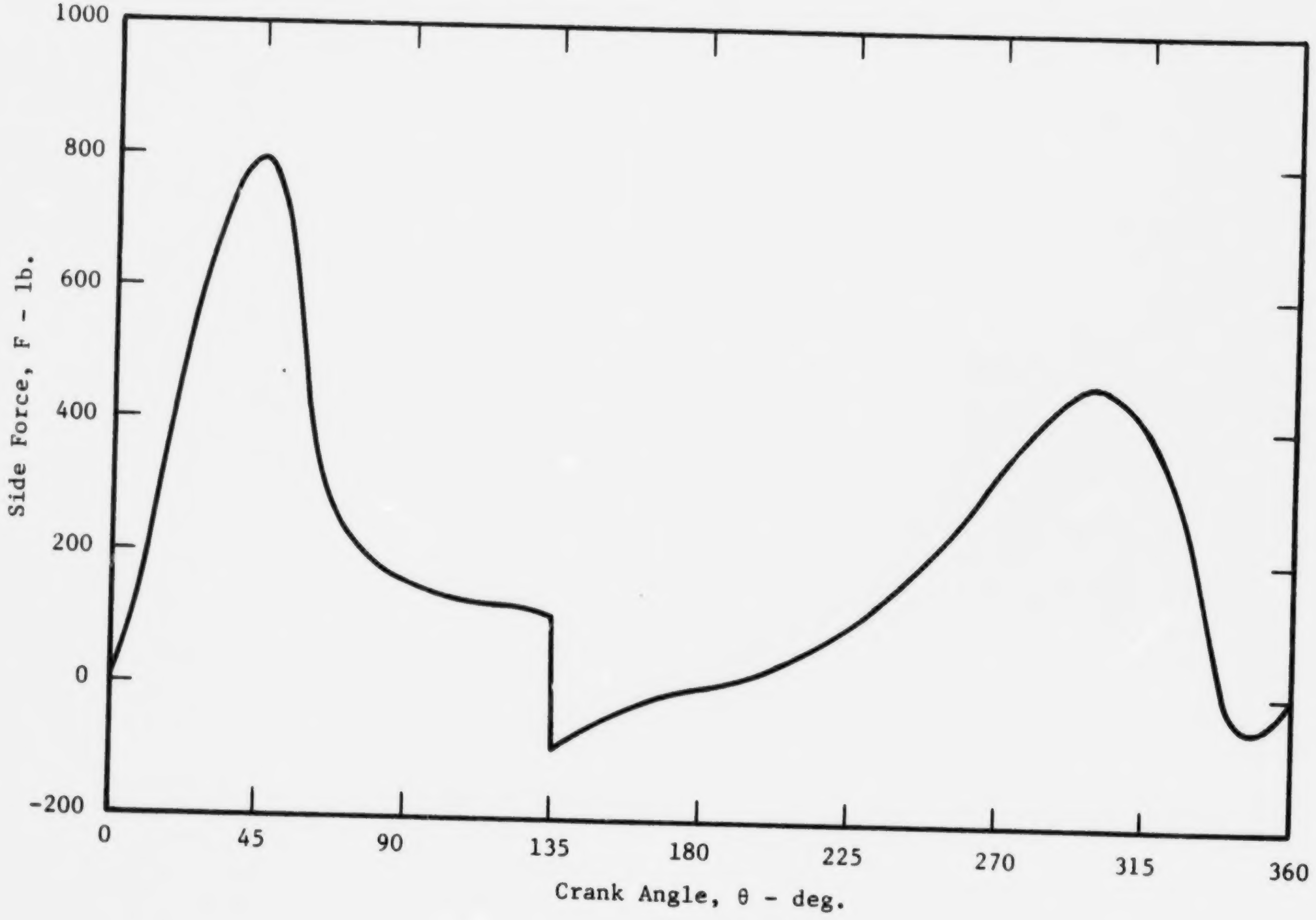


Figure 2-16. Wrist Pin Side Force Due to Pressure Force, Reciprocating Weight Inertia Force at 2500 RPM, and Inertia Force Due to Regular Acceleration of Connecting Rod - Crosshead Expander.

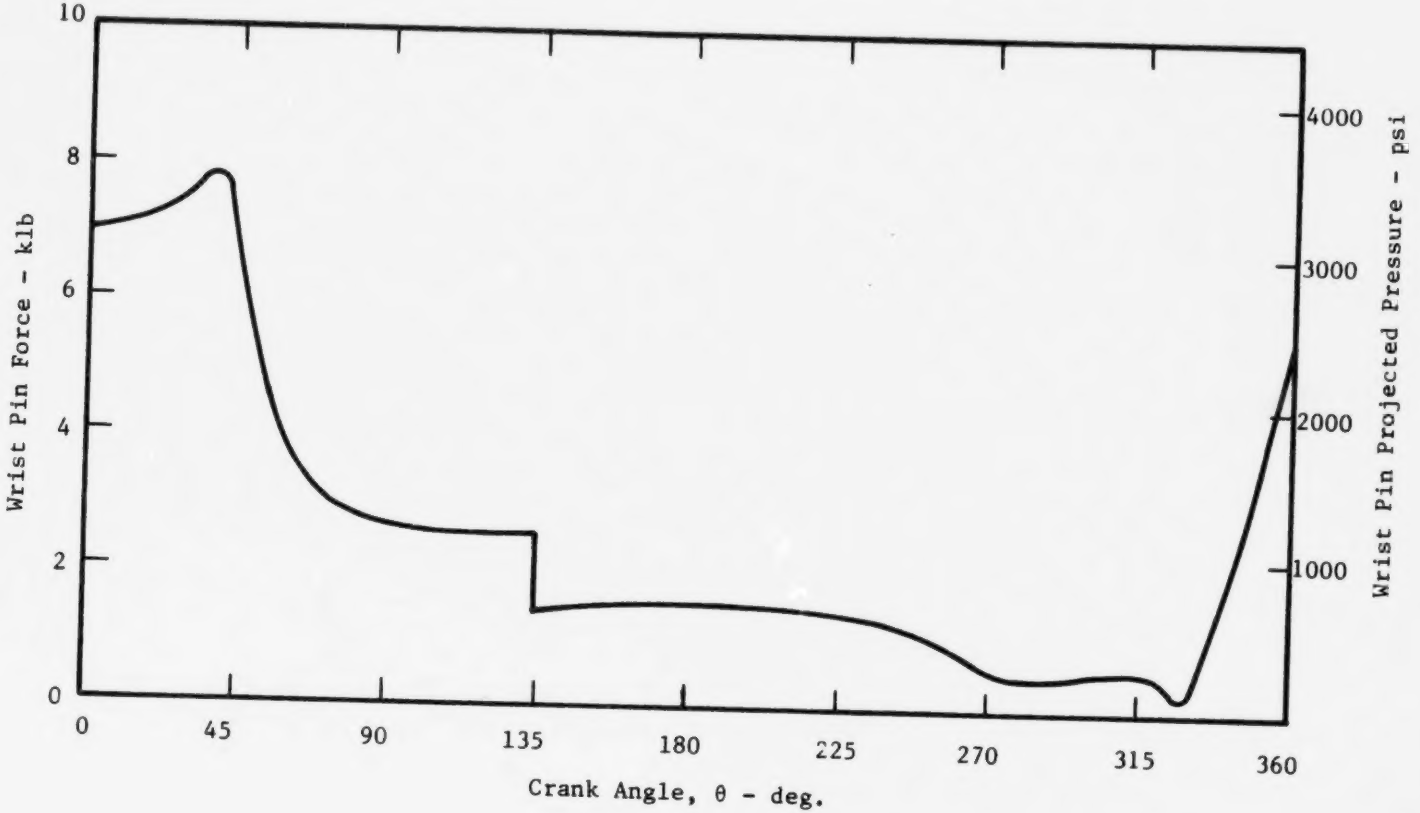


Figure 2-17. Crosshead Vertical and Side Forces and Projected Pressure on Wrist Pin at 2500 RPM - Crosshead Expander.

Table 2-2 and Figure 2-18 (Drawing 707E728) identify the stresses and factors of safety at the principal points of interest in the crosshead expander.

One of the high stress areas of the piston head is at location A (refer to Figure 2-18), the bottom of the retaining nut hole. Assuming that the alternating forces are producing pure shear stresses and using the allowable alternating stress of the material for 1000 hr. at 600°F, the factor of safety (FS) is 3.41 against fatigue failure. This is satisfactory even if the stress concentration factor (SCF) in this region were 3.

The junction of the crown and walls of the piston (loc. B) is difficult to analyze exactly with standard formulae because of the rapid changes in cross-section from crown to groove to land, but very generous FS exist even assuming that the inlet pressure leaks past the top ring.

The land bending FS = 13.4 (loc. C), and the land deflection is only 0.06 mils. With a clearance of 0.002 in. between the lands and the rings, no binding of the piston rings will occur.

The generous fillet and taper beneath the piston head will minimize the stress concentrations applicable to the pressure forces originating in the crown and lands.

The alternating bearing stresses between the piston head and the piston rod (loc. E) are satisfactory.

The most critical problem for the piston nut and the threads of the piston rod (loc. F) is that of motoring at high speed. During motoring, the nut must absorb the crown inertia forces, while the pressure forces will unload the nut. With a specified preload force and using SCF = 3.85 for thread fatigue, the thread FS = 1.25 for motoring at 2500 RPM.

The two power piston heads are designed to have the same assembly weight. The decreased groove stock of the piston head with metallic rings was placed so as to increase its safety.

Table 2-2

Stresses and Factors of Safety at Principal Points of Interest in the Crosshead Expander (Figure 2-18 Identifies Location)

Location	Effective Stress Cycle, ksi	Allowable(a) Stress, ksi	Factor of Safety, -
A	12.3 ± 12.3	± 42.0	3.41
B	0.9 ± 0.9	± 42.0	47.3
C	3.1 ± 3.1	± 42.0	13.4
E	22.6 ± 13.2	± 47.0	3.57
F	14.2 ± 52.0	± 64.8	1.25
G	20.5 ± 20.5	± 50.0	2.44
H	2.7 ± 0.5	± 4.5	9.09
I	44.7 ± 12.1	± 48.8	4.05
J	2.1 ± 2.1	± 4.5	2.14
K	36.5 ± 24.3	± 61.8	2.55
L	86.0 ± 15.6	± 31.5	2.03
M	17.6	43.0	2.44
N	42.4 ± 42.4	± 75.0	1.77
O	17.7 ± 17.7	± 42.0	2.37
P	78.0 - 88.1	125.0	1.42(b)
Q	12.2 ± 10.3	± 51.5	5.00
R	42.6 ± 7.6	± 40.0	5.26
S	75.2 ± 29.1	± 55.0	1.88
T	30.1 ± 13.7	± 34.0	2.48
U	9.6 ± 0.8	± 50.5	11.63
V	53.0 ± 20.5	± 31.0	1.51
W	57.3 ± 8.2	± 35.0	4.27
X	67.8	81.0	1.19
Y	1.1 ± 1.1	± 18.5	17.1
Z	30.7 ± 27.3	± 44.8	1.64
AA	41.6	84.0	2.02
AB	43.9	120.0	2.82
AC	27.0	32.0	1.19
AD	5.5 ± 5.5	± 16.0	2.94

Table 2-2 (Cont'd.)

Stresses and Factors of Safety at Principal Points of Interest in the Crosshead Expander (Figure 2-18 Identifies Location)

Location	Effective Stress Cycle, ksi	Allowable[(a)] Stress, ksi	Factor of Safety, -
AE	12.3	32.0	2.60
AF	2460 lb	3590 lb	1.46[(c)]
AG	10.8 ± 10.8	± 20.0	1.86
AH	57.7 ± 20.9	± 44.5	2.13
AI	77.2 ± 16.8	± 36.0	2.14
AJ	15.1 ± 15.1	± 20.0	1.33

NOTES: (a) The allowable stresses shown as ± are the allowable alternating stresses when the mean stress if the same as shown in the effective stress tabulation.

(b) Spring - Maximum allowable stress is 125.0 ksi for initial stress of 78.0 ksi. Maximum imposed stress is 88.1 ksi.

(c) FS = critical load/applied load.

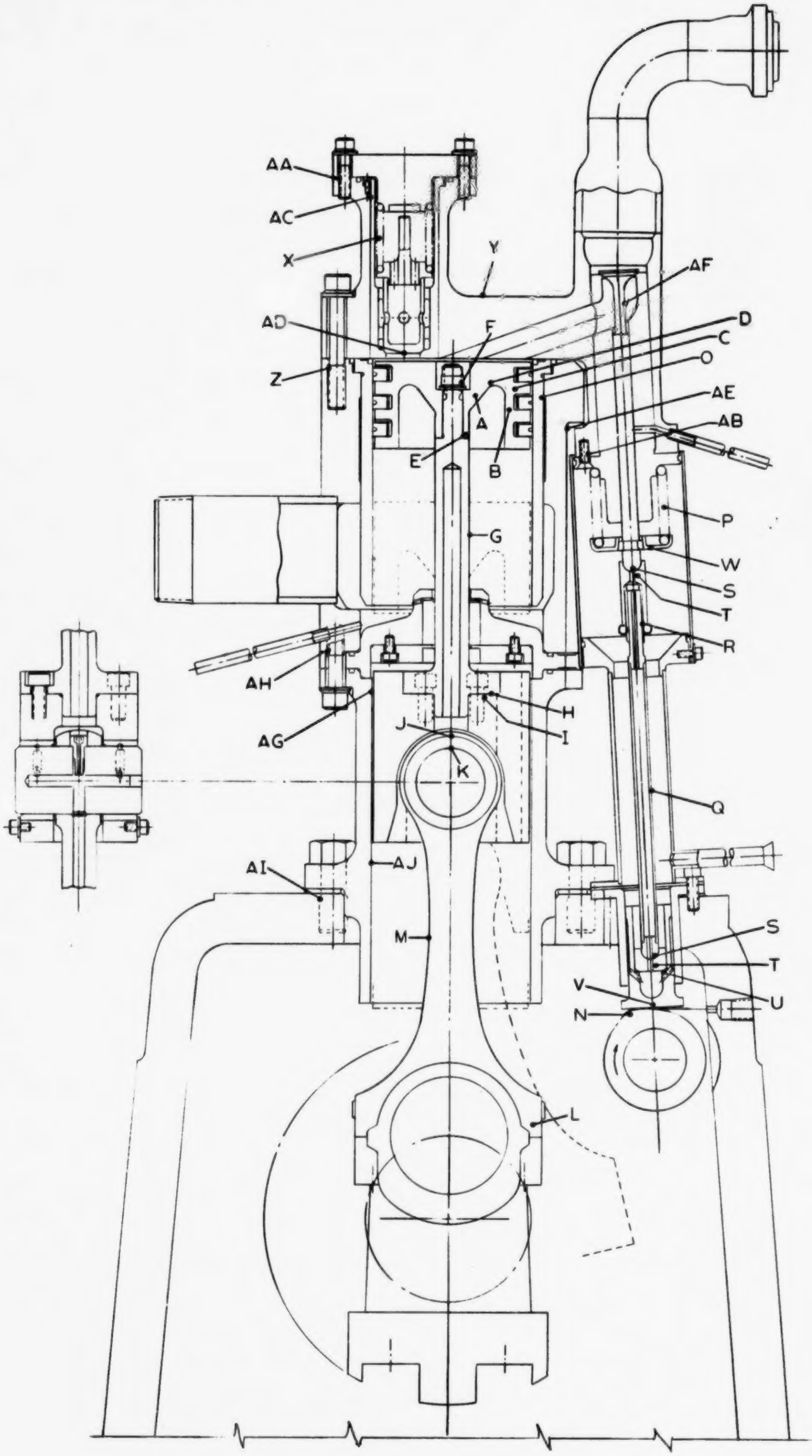

AA
AC
X
Y
AF
AD
F
D
C
O
Z
AE
AB
A
E
B
P
G
W
S
T
R
AH
H
AG
I
J
K
Q
AI
AJ
M
S
T
V
U
N
L

Figure 2-18. Location of Stresses, Crosshead Expander (GE Dwg. 707E728).

Using the standard secant formula for short-column buckling, the piston rod FS = 2.44 (loc. G) against buckling.

The bearing load between the rod end and the top of the aluminum crosshead piston (loc. H) produces FS = 4.79 vs. fatigue failure.

With a selected preload force, the bolts joining the rod and piston (loc. I) had FS = 4.05. Even though FS = 0.80 for assuring that the bolts break before the aluminum threads strip, still FS = 3.24 against the tapped threads failing. A thread fatigue SCF = 3 is included in the above calculations.

The projected wrist pin bearing pressures (loc. J) are compressive in one direction during the power stroke and compressive in both vertical directions during motoring. Neither exceed 4000 psi and are acceptable. The stresses due to the inertia loads of the crosshead skirt beneath the wrist pin are negligible.

For 1000 hr. fatigue loading at 300°F, FS = 2.55 for the wrist pin with SCF = 3 included because of the oil hole at the maximum stress point (loc. K).

The connecting rod bolts (loc. L) have FS = 2.03 for 1000 hr. operation at 300°F, with a fatigue SCF = 3.85 assumed.

Against short-column buckling, the connecting rod FS = 2.44 (loc. M). Against alternating axial and bending stresses (including the whipping effect) the FS = 3.64.

The original manufacturer's crankshaft design is being used, and it is understood that the crankshaft design has withstood pressure loads and speeds similar to those planned for significant periods of time. No crankshaft stress analyses are planned.

The maximum bending stress in the camshaft is less than 800 psi, unbalance loads included, and is satisfactory. The cam Hertzian stress FS = 1.77 vs. fatigue failure.

The region of maximum interest in the cylinder block liner is at loc. O where the cylinder support changes from complete restraint by the block to completely unrestrained. One liner will be made of 440C stainless steel; its replacement of 1 Cr-1/2 Mo-1/4 steel. The liners will be shrunk fit at assembly so that line contact still exists at the 600°F operating temperature. For 440C the assembly FS = 5.12, and for 1 Cr-1/2 Mo-1/4 the assembly FS = 78 against room temperature yielding.

During operation, the combined effect of pressure forces, an assumed ΔT = 200°F across the cylinder wall, and the change in restraint produces FS = 4.57 in the 440C liner and FS = 2.37 in the 1 Cr-1/2 Mo-1/4 liner against 1000 hr. fatigue at 600°F.

A compression spring (loc. P) of standard size and material was chosen which provided a preload of 177 lb. and a maximum force of 200 lb. at 0.100 in. cam lift. The corresponding alternating stresses of 78.0 - 88.1 ksi provided FS = 1.42 against fatigue. Its natural frequency of 17,095 cpm was satisfactory for the first six harmonics of a 2500 RPM driving force. Buckling is not a problem.

All of the valve train components (loc. Q-W) were analyzed with respect to fatigue and had FS = 1.51 - 11.6. The pushrod RS = 2.28 against buckling.

The cam profile prevents impact loading of the inlet valve sealing faces.

A standard size compression spring (loc. X) was chosen for loading the recompression valve. Inconel X-750 material was specified because the spring will be continually soaked in the 900°F inlet steam. After an assumed three percent relaxation, the spring preload will be 78 lb. Since this valve is designed to operate only occasionally, fatigue is not a problem and the corresponding stress of 67.8 ksi produces a FS = 1.19 vs. the proportional limit of the material at 900°F. The first critical frequency of 18,150 cpm was satisfactory for the first seven hormonics of 2500 RPM. Buckling is not a problem.

The selected preload of the spring sets the valve lift-off pressure at 214 psia during motoring at ambient pressure, and 1200 psia when the inlet pressure is 1000 psia. The 1200 psia opening pressure is 447 psi less than that required to open the steam inlet valve at 1000 psia inlet pressure. The 214 psia opening pressure is 193 psi less than that required to open the inlet valve at 14.7 psia inlet pressure.

The basic inlet manifold head FS = 17.1 (loc. Y) without consideration of the SCF of holes, ports, and instrumentation fittings. None of these are expected to produce unacceptable conditions.

The head bolts (loc. Z) and all others in the expanders are provided with a preload which gives FS = 1.25 against the opening of their joints by pressure forces. The head bolt FS = 1.64 against fatigue, with SCF = 3.85 assumed.

The recompression valve and inlet valve housing bolts are subjected only to steady loads. The recompression valve bolt (loc. AB) FS = 2.02 vs. yielding at 900°F and SCF = 1. Some yielding can occur if the same bolts are used for 1000 hr., since the applied stress exceeds the allowable 1000 hr. creep strength of the bolt material at 900°F.

The inlet valve body screw (loc. AB) FS = 2.82 against yielding at 300°F with SCF = 3.

The recompression valve body (loc. AC) has a minimum FS = 1.19 against rupture in 1000 hr. at 900°F, without consideration of the generous fillet radius at this location which will increase its safety.

The recompression valve (loc. AD) FS = 2.94 against 1000 hr. fatigue at 900°F.

The upper cylinder block (loc. AE) FS = 2.60 against 1000 hr. rupture, assuming full axial loads and 1000 psia inside.

The intake valve stem (loc. AF) FS = 1.46 against buckling.

The midflange cylinder/flange intersection of the lower cylinder block (loc. AG) FS = 1.86 against 1000 hr. fatigue.

The midflange bolts (loc. AH) FS = 2.13 against 1000 hr. fatigue with SCF = 3 included.

The base bolts (loc. AI) FS = 2.14 against 1000 hr. fatigue with SCF = 3 included.

The lower cylinder/flange intersection of the lower cylinder block (loc. AJ) FS = 1.33 against 1000 hr. fatigue.

Two counterrotating shafts with counterweights are included in the Waukesha crankcase to balance the primary (1/rev) component of the inertia force due to the reciprocating parts of the engine. A secondary force component (2/rev), which is smaller, will not be balanced. The reciprocating masses include the power and crosshead pistons, the piston rod, 1/3 of the weight of the connecting rod shank, and the upper end of the connecting rod with wrist pin and bearings. These weights are tabulated in Table 2-3.

The existing counterweights in the Waukesha crankcase do not contain sufficient mass, and geometrical limitations prevent the addition of heavier masses to the existing counterweights. However, high density sintered material with high tungsten content (Mallory 1000) will provide the proper balancing mass within the geometrical limitations. Stress levels in the counterweights and their bolts are 2.3 ksi and 46.2 ksi at 2500 RPM. Their respective factors of safety are 32 and 2.88 against fracture at 300°F.

The crankshaft counterweights balance the expander rotating components. Since the Waukesha crankshaft is already balanced for a specific rotating weight, the existing balancing weight need only be adjusted to compensate for the difference between the rotating mass of the Waukesha engine and that of the steam expander. The existing bolts are to be used, with small amounts of additional weight added beneath their heads. The rotating weights of the two expanders are listed in Table 2-3.

2.4.3.2 Trunk Expander

There are virtually no differences between many of the components of the trunk expander and the corresponding ones of the crosshead expander. The principal changes are in the piston assembly and the connecting rod.

Table 2-3

Reciprocating and Rotating Weights

	Reciprocating	
	Crosshead Expander	Trunk Expander
Power piston head	2.996 lb.	5.683 lb.
Power piston rod	1.445	--
Crosshead piston	1.434	--
Crosshead bushing (2)	0.271	0.271
Wrist pin	1.359	1.359
Connecting rod	0.748	0.780
Total	8.253 lb.	8.093 lb.

	Rotating	
	Crosshead Expander	Trunk Expander
Connecting rod	3.238 lb.	3.319 lb.
Con. rod bearing	0.254	0.254
Total	3.492 lb.	3.574 lb.

The piston assembly consists of a steel head (upper part) and an aluminum skirt (lower part). When the aluminum skirt and steel head are assembled with line-to-line contact in the region of the rider ring groove, the greater thermal expansion of the aluminum will produce a high contact pressure at elevated temperatures. Compressive stresses are developed in the aluminum, while tensile stresses are developed in the steel. The lower strength of the aluminum makes it the controlling part. On the basis of the 1000 hr. tensile rupture strength of aluminum, the maximum allowable operating temperature of the aluminum skirt is 450°F if both parts are at the same temperature and were line-to-line fit at assembly. If the thermal analysis indicates a higher operating temperature, the controlling diameter will have a loose fit at assembly.

By Assuming conservatively that the aluminum temperature could reach 450°F during motoring, the combination of thermal load, bolt preload, and inertia load produces $FS = \frac{84.1 \pm 28.5 \text{ ksi}}{84.1 \pm 14.7 \text{ ksi}} = 1.94$ against 1000 hr. bolt fatigue. In order to assure that the bolts break before the aluminum threads strip, the bolt needs to penetrate 1.54 in. into the aluminum skirt. This engagement length cannot be achieved within the geometrical limitations, and the 1 in. engagement length which is possible gives $FS = 1/1.54 = 0.65$ that the bolts break before the tapped threads strip. The tapped threads are still satisfactory, since $FS = 0.65\ (1.94) = 1.26$ against the tapped threads stripping in 1000 hr. at 450°F. All motoring will be for much shorter periods, and the pressure loads of normal power operation will decrease the forces on the bolts and threads.

The combination of pressure load, bolt preload, inertia load, and thermal load due to the piston head being 100°F hotter than the bolts gives $FS = 1.67$ against bearing fatigue of the aluminum crown interface with the steel head.

As was the case for the crosshead power piston, simple formulae do not give exact answers for the trunk power piston stresses because of the abrupt changes in geometry. However, the minimum factor of safety in the vicinity of the crown/cylinder junction is at least $\frac{24.2 \pm 45.5 \text{ ksi}}{24.2 \pm 24.2 \text{ ksi}} = 1.88$ against 1000 hr. fatigue at 600°F. The safety will be higher because the generous fillet in this region was not considered and the average cylinder thickness will be larger than that used to calculate the above stresses.

Against short-column buckling, the connecting rod FS = $\frac{43.0 \text{ ksi}}{19.1 \text{ ksi}} = 2.25$. Against alternating axial and bending stresses (including the whipping effect) the FS = $\frac{-15.5 \pm 54.0 \text{ ksi}}{-15.5 \pm 15.5 \text{ ksi}} = 3.49$.

Further analysis of the trunk expander will be reported in the future.

2.4.4 Piston Ring Wear Prediction

Zero wear is defined as wear so slight that the surface finish of the wear track is not significantly different from the finish of the virgin surface(4). The wear scar is, then, roughly equal to one-half of the peak-to-peak value of the surface finish.

The engineering model states that the wear between two loaded sliding surfaces can be controlled by limiting the maximum shear stress, τ_{max}, in the contact region. Zero wear can be obtained for a specific number of passes if:

$$\tau_{max} \leq \gamma \, \tau_y \qquad \text{psi}$$

The fractional parameter γ is dependent upon the number of passes, the materials and lubricant used, and the lubricating conditions, as shown in Table 2-4. For 2000 passes, γ is designated as γ_R.

Table 2-4

Fractional Parameter

Type of Lubrication	γ_R	Material/Lubricant System
Quasi-hydrodynamic	1.00	Fully hydrodynamic lubrication
	0.54	Partial hydrodynamic lubrication
Dry or boundary lubrication	0.54	Low susceptibility to transfer
	0.20	High susceptibility to transfer

Since no published data exists for the exact material combinations to be used in the single cylinder expanders, it is estimated that $\gamma_R = 0.20$ for the unlubricated graphite ring in contact with a steel cylinder. This estimate is conservative, and is the value found for dry 52100 steel vs. 7 stainless steels, vs. 19 of 20 steels, vs. 3 nickel alloys, vs. 3 copper alloys, vs. 6 aluminum alloys, vs. 1 of 2 sintered brasses, vs. 2 sintered bronzes, vs. 1 of 3 sintered irons, vs. 2 sintered iron-copper alloys, and vs. 3 sintered steels.(2)

The properties of the piston ring material, U.S. Graphite Co. Grade 107, are shown in Table 2-5. In the absence of specific information concerning its allowable shear yield strength, it is assumed that $\tau_y = 7500$ psi, the RT tensile strength. The ring material is considered so brittle that the yield and fracture strengths are equal. (Another method of estimating τ_y from the microhardness gives a larger allowable τ_y.)

Table 2-5

Properties of Grade 107 Carbon Graphite

Composition - Carbon graphite impregnated with Sb

Strength - at RT, will increase with temperature

	Compression	- 38,000 psi
	Transverse fracture	- 10,000 psi
	Tensile	- 7,500 psi

Elastic Modulus - $E = 3.5 \times 10^6$ psi

Hardness - Sclersocope 92
DPH (converted) 865

The pressure, and therefore, frictional loads on the ring/cylinder interface will not be constant throughout the cycle. The pressure differential will decrease toward zero as the steam expands in the cylinder and the exhaust ports are uncovered. Using an idealized representation of the pressure vs. crank angle relationship such as shown by Figure 2-14, a mean effective cylinder pressure of 290 psi will occur for the 15% cut off condition.

The contact load, F' at the ring/cylinder interface will be a function of the spring load, F'_s, the pressure load, F'_p, the frictional load, F'_u, and the ring geometry as shown by Figure 2-19.

Assuming that all radially outward adjustments in ring position which are required for wear compensation are made at bottom dead center, where all forces are zero except for the spring force, the horizontal frictional force is zero during the rest of the cycle. The contact stress equation reduces to:

$$q_o = \frac{F'}{A'} = \frac{4 + 0.0622\ \Delta P - 0.0296\ P_1}{0.2415} \qquad \text{psi}$$

To determine conditions which will ensure zero wear for a desired life of N passes for a given mechanism, the following relation has been derived:

$$\tau_{max} = K\ q_o \sqrt{0.5^2 + \mu^2} \leq \left(\frac{2000}{N}\right)^{1/9} \gamma_R \tau_y \qquad \text{psi}$$

Solutions are shown in Figure 2-20 for the graphite piston rings, for the average value of P_1 = 290 psia, ΔP equal to three fractions of P_1, and coefficients of friction from 0.05 to 0.50 as functions of rotative speed.

By combining equations the fractional area of the piston ring with a stress concentration less than the allowable for zero wear in 1000 hours is determined. These calculations are also plotted in Figure 2-20 as fractional area with zero wear. The significance of this figure is to show that, according to analytical predictions, the graphite piston rings should experience only minor wear near the edges under the ideal conditions assumed in the calculations.

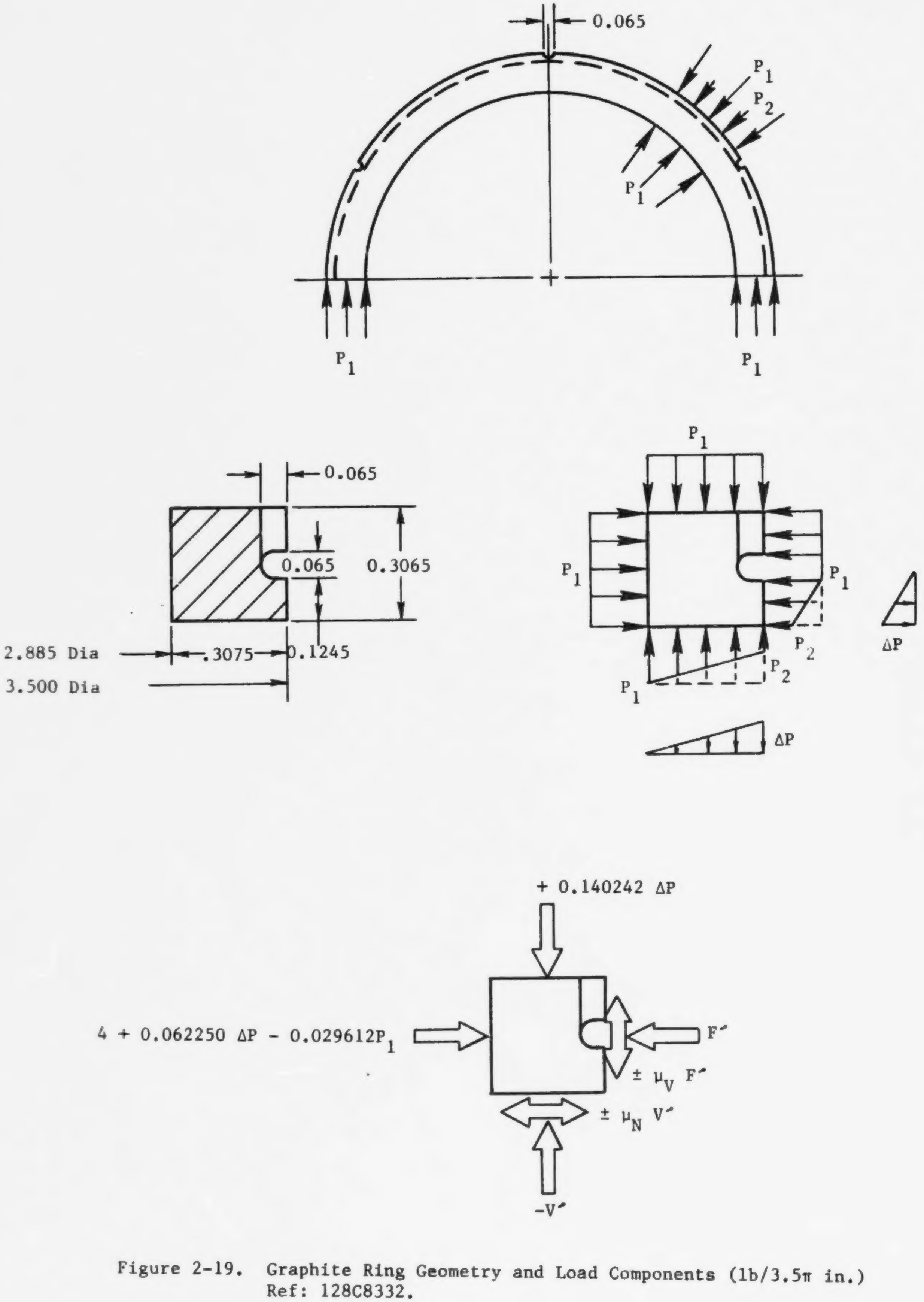

0.065
P_1
P_2
P_1
P_1
P_1
0.065
0.065
0.3065
2.885 Dia
.3075
0.1245
3.500 Dia
P_1
P_1
P_1
P_2
P_2
P_1
ΔP
ΔP
+ 0.140242 ΔP
4 + 0.062250 ΔP - 0.029612P_1
F′
± μ_V F′
± μ_N V′
-V′

Figure 2-19. Graphite Ring Geometry and Load Components (lb/3.5π in.)
Ref: 128C8332.

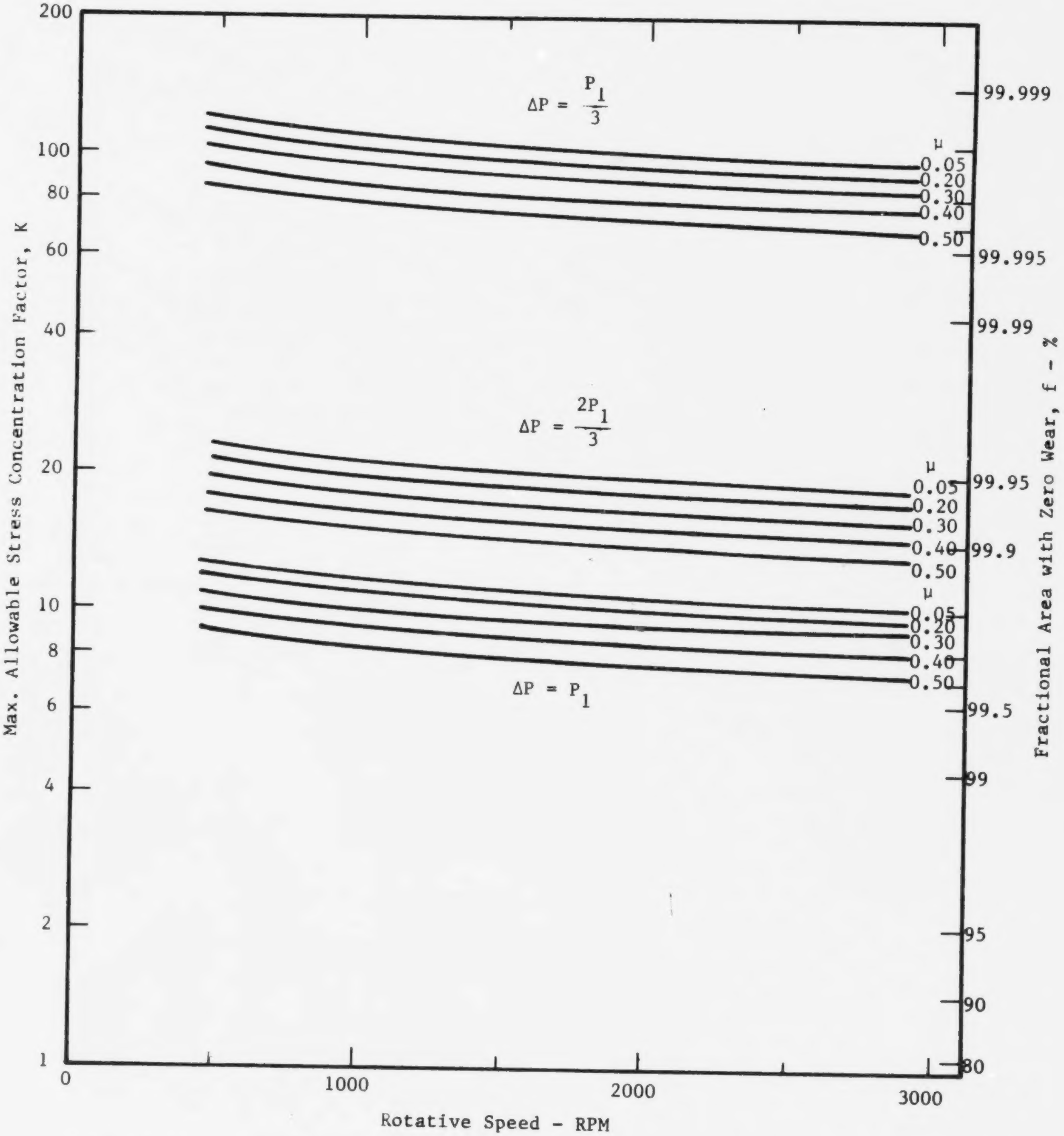


Figure 2-20. Maximum Allowable Stress Concentration Factor and Fractional Area with Zero Wear of Graphite Piston Rings in 1000 Hr.
Mean P_1 = 290 psia = upstream pressure on piston ring.
ΔP = pressure drop across piston ring.

3.0 TASK II EXPANDER MATERIALS

3.1 Expander Materials Technology

In recently designed steam reciprocating engines for automotive applications, maximum steam temperatures of about 700°F have been utilized. Temperatures have been restricted to this general range by materials limitations, in one form or another. For example, the GM SE-101 engine had a maximum steam temperature of 700°F in recognition of the possible thermal degradation of the organic lubricant used to prevent excessive wear at the steam piston-wall interface. A design pressure of800 psia arose from considerations of expander bearing loads. These operating conditions (700°F, 800 psia) might be considered to be current practice in the design of steam engines for automotive applications. Engine thermal efficiency is quite limited, under these conditions, as compared with modern central steam power stations operating at 1000° to 1050°F and 2400 psia. These advanced conditions have been reached by increasing technology over the last 40 years, the same period in which there has been essentially no increase in the cycle conditions for the steam car engine.

The higher temperatures associated with the more efficient automotive steam engine place constraints on materials due to mechanical strength and corrosion considerations. To insure the long life (∿ 3000 hours) required for satisfactory use in the reciprocating steam expanders, the materials must be resistant to oxidation, corrosion by steam and have adequate mechanical properties (yield strength, resistance to creep and cyclic loading, hardness, etc.) at the anticipated operating conditions. Surface properties can be favorably altered by the application of a surface coating; use of surface coatings are considered where appropriate, i.e., wear surfaces. In addition to the constraints imposed on materials by the engine operating conditions, economic constraints can limit the utilization of existing technology in the solution to materials problems.

Cost is a major factor in large quantity production of automotive components. Economy dictates that the lowest cost alloy or material, with properties suitable to the service conditions, be used.

Probably, the most limiting factor in the successful operation of highly efficient steam engines will be lubrication. The dry conditions which superheated steam imposes on a system at the high operating conditions can cause severe galling and excessive wear problems in those components whose surfaces are in relative motion, i.e., the cylinder liner/piston ring interface and the inlet valve face/seat and inlet valve stem/guide interfaces. Lubrication of these components are discussed in Section 4.0.

A literature search and industrial survey was made to identify and evaluate the current materials technology in support of the expander designs described in Section 2.0. Personal visits were made to the Ford Motor Company Automotive Research and Engineering Center and the General Electric Company Diesel Engine Department, Large Steam Turbine and Generator Department and Medium Steam Turbine and Generator Department to identify materials used in current reciprocating and steam turbine components. To supplement the information obtained from these visits, library searches were made covering automotive components and oxidation and wear resistant materials for use in dry steam environment through the General Electric Technical Information Center Library and Automatic Information Retrieval System, the SAE and AIME-ASM Transactions and the Engineering Index.

A summary of the results obtained from the technology review follows:

3.1.1 Cylinder Block, Cylinder Head, Intake Manifold, Exhaust Manifold

Experience with materials in large central station steam power station has shown that the useful life of steel is essentially unlimited if the proper steel is selected and the service conditions are properly controlled. Some steels have been in service for 40-50 years. Invariably when a failure occurs the cause can be attributed to impurities in the steam and the problem usually is solved by correction of the service conditions. Plain carbon and low alloy steels are commonly used for the fabrication of pressure-containing components such as feedwater piping, steam piping, valve bodies, flanges, discs, etc. that are regularly found in steam generating systems. The most commonly used steels are listed in Table 3-1 together with the appropriate ASTM designations and maximum service temperatures.

The use of these steels up to the maximum temperature listed in Table 3-1 will vary with the anticipated service conditions and from user to user. For example, one manufacturer limits the use of 1.25% Cr-0.5% Mo steel to 800°F. In actual practice carbon steels have been used successfully in steam service at temperatures to 750°F for 25 to 30 years. Similarly, the 0.5% Mo, the 1.25% Cr-0.5% Mo and the 2.25% Cr - 1.0% Mo alloy steels have proven to be satisfactory for service at temperatures up to 850°, 1000° and 1050°F respectively. However, there is a major restriction in the use of 0.5% Mo steel in that is is not recommended for use in welded structures. Welded 0.5% Mo steel has a strong tendency to graphitize in high temperature service. All of these steels can be considered for use in the major static components for reciprocating automotive steam expanders.

In contrast to the pressure-containing components in steam generating systems, the materials currently being utilized in the production of the cylinder blocks, cylinder heads and intake manifolds of gasoline and diesel fueled internal combustion engines for passenger cars, trucks and locomotives are gray cast iron and cast steel. The most common gray cast iron being used for the production of these components in passenger cars is SAE G4000 (ultimate strength - 40,000 psi). SAE G4000 contains 3 - 3.3% C, 1.8 - 2.1% Si, 0.6 - 0.9% Mn and consists of a lamellar pearlite matrix with Type A flake graphite. For parts subjected to higher pressure or heavy duty engines, SAE G6000 (ultimate strength 60,000 psi) gray cast iron is used. In one large diesel engine the cylinder head is produced from a cast, weldable Mn-Mo steel (0.20% C - 1.35% Mn - 0.25% Mo). The steel casting is normalized at 1650°F (during the hardening of the valve seats) to a hardness of 180 - 225 BHN. This treatment produces an ultimate strength of 90,000 psi, a yield strength of 60,000 psi, tensile elongation of 10% and reduction-in-area of 25%. Special heat resistant parts are frequently made from Ni-Mo alloy steel. Intake manifolds in some internal combustion engines are also being produced from cast aluminum alloy 355

Table 3-1

Plain Carbon and Low Alloy Steels Used for Pressure Containing Components in Steam Generating Plants(a)

	Alloy	ASTM Designation	Form	Max. Service Temp., °F
1.	Carbon Steel (0.35% C max)	A216 (Grade WCB)	Casting	850
2.	Carbon Steel (0.25% C max)	A216 (Grade WCA)	Casting	850
3.	Carbon Steel (0.35% C max)	A105 (Grades 1, 11)	Forging	850
4.	Carbon Steel (0.25-35% C max)	A106 (Grades A, B,C)	Pipe	850
5.	Carbon Steel (0.06-0.18% C - 0.35% Si max)	A192	Boiler Tube - High Pressure	850
6.	Carbon Steel (0.27% C max)	A210	Superheater Tube	850
7.	0.25% C - 0.5% Mo	A217 (Grade WC1)	Casting	850
8.	0.25% C - 0.5% Mo	A182 (Grade F1)	Forging	850
9.	0.15% C - 0.5% Mo	A335 (Grade P1)	Pipe	850
10.	0.20% C - 0.5% Mo	A209 (Grade Tia)	Boiler and Superheater Tubes	950
11.	0.15% - 1.0% Cr - 0.5% Mo	A182 (Grade F12)	Forging	1050
12.	0.15% C - 1.0% Cr - 0.5% Mo	A225 (Grade P12)	Pipe	1000
13.	0.20% C - 1.25% Cr - 0.5% Mo	A217 (Grade WC6)	Casting	1050
14.	0.15% C - 1.25% Cr - 0.5% Mo	A182 (Grade F11)	Forging	1050
15.	0.15% C - 1.25% Cr - 0.5% Mo	A225 (Grade P11)	Pipe	1050
16.	0.15% C - 1.25% Cr - 0.5% Mo	A213 (Grade T11)	Boiler & Superheater Tubes	1100
17.	0.18% C - 2.5% Cr - 1.0% Mo	A217 (Grade WC9)	Casting	1050
18.	0.15% C - 2.25% Cr - 1.0% Mo	A182 (Grade F22)	Forging	1050

Table 3-1 (Cont'd.)

Plain Carbon and Low Alloy Steels Used for Pressure Containing Components in Steam Generating Plants[a]

	Alloy	ASTM Designation	Form	Max. Service Temp., °F
19.	0.15% C - 2.25% Cr - 1.0% Mo	A225 (Grade P22)	Pipe	1050
20.	0.15% C - 2.0% Cr - 0.5% Mo	A213 (Grade T3b)	Boiler & Superheater Tubes	1200
21.	0.15% C - 2.25% Cr - 1.0% Mo	A213 (Grade T22)	Boiler & Superheater Tubes	1200
22.	0.15% C - 3.0% Cr - 1.0% Mo	A213 (Grade T21)	Boiler & Superheater Tubes	1200
23.	0.15% C - 5.0% Cr - 0.5% Mo	A213 (Grade T5)	Boiler & Superheater Tubes	1200
24.	0.15% C - 9.0% Cr - 1.0% Mo	A213 (Grade T9)	Boiler & Superheater Tubes	1200
25.	Type 304 SS (18% Cr - 10% Ni)	A213 (Grade TP304)	Boiler & Superheater Tubes	1500
26.	Type 321 SS (18% Cr - 10% Ni-Ti)	A213 (Grade TP321)	Boiler & Superheater Tubes	1500

(a) Reference 5.

(Al-5% Si - 1.25% Cu - 0.5% Mg - 0.1% Zn - 0.2% Ti - 0.1% Mn - 0.2% Fe).

Materials used for the exhaust manifold in internal combustion engines are subjected to the highest temperatures in the engine and would correspond to the intake manifold of the steam expander. Alloyed cast iron is most commonly used for the exhaust manifold in the internal combustion engine. The SAE G4000d, a Cr-Mo alloyed cast iron and SAE G4000e a Cr-Mo-Ni alloyed cast iron are typical. The range of operating temperatures that have been reported for the higher temperature components of the internal combustion engine that require the use of alloyed gray cast iron are:

Component	Operating Temperature Range, °F
Cylinder Heads	450 - 1000 (locally)
Exhaust Manifolds	300 - 1200
Exhaust Valve Seat	800 - 1300

The alloyed cast irons can be considered for components of the steam expander that operate at lower temperatures and pressures. Where strength is critical and greater reliability (safety) and oxidation resistance is required, as is encountered in high pressure steam systems, nodular cast iron is utilized. Typical nodular cast irons are SAE D5506, D4512 and D7003. SAE D5506 has a yield strength (0.2% offset) of 55,000 psi and a tensile elongation (∿ 2 inches) of 6%; the structure is ferritic-pealitic. SAE D4512 has a yield strength of 45,000 psi, a tensile elongation of 12% and has a ferritic structure. SAE D7003 has a yield strength of 70,000 psi, a tensile elongation of 3% and a pearlitic structure.

3.1.2 Cylinder Liners/Piston Rings

As stated previously, the high-speed relative motion between the cylinder liner and the power piston rings in combination with the high temperatures and pressures that exist within the cylinder chamber make these components two of the most critical components in the engine. Although the materials currently in use as cylinder liners and piston rings in gasoline and diesel fueled internal combustion engines were reviewed, the technology is not directly applicable to the reciprocating steam expanders being designed to operate with dry solid lubricants. However, cylinder liner/piston ring material combinations used in oil free air compressors are applicable and will be discussed in Section 4.0.

The materials most commonly used for cylinder liners under oil lubricated conditions are alloyed gray cast iron, the Meehanite cast irons, chromium plated gray cast iron and nitrided gray cast iron. Typical of the alloyed gray cast iron is a composition of 3.35% C - 2.15% Si - 0.65% Mn - 0.4% Cr; the liner is hardened by quenching from 1600°F into salt at 475°F followed by cooling in air. The resulting hardness is Rockwell C 45 minimum. The Meehanite cast irons are high strength, fine grained castings combining the properties of cast iron and steel. A commonly used Meehanite, Grade GA, has an ultimate strength of 50,000 psi. Liners that are chromium plated or nitrided are produced from centrifugally cast gray cast iron. The liners are centrifugal cast in order to achieve the bore surface integrity (minimum porosity) that is required for the hard chromium plating and nitriding processes. A typical centrifugally cast gray cast iron used in liners for large diesel engines has an ultimate strength of 35,000 psi and a hardness of ∿ 250 BHN.

Gray cast iron also is the most common material used for oil lubricated compression and oil piston rings. Occasionally AISI 1070 and 52100 bearing steels are used. However, the steel segment or steel rail types of oil rings are almost always made of AISI 1070 to 1095 steels. The chromium plating of piston rings has been found to reduce wear of the rings and cylinders by ∿ 75%. However, chromium cannot be run against itself so that the rings cannot be chromium plated if the liner is chromium plated. Chromium plating is used primarily on compression rings in heavy duty engines and on oil rings of the segmented type. Since chromium plate reduces the fatigue endurance limit of cast iron, higher strength cast irons must be specified for chromium plating applications. Two examples of cylinder liner/piston ring combinations are shown in Table 3-2.

The K-28 nodular cast iron (3.3% - 2.2% Si - 0.5% Mo - 0.5% Cu - 0.05% Mg) is a centrifugally cast martensitic ductile iron and has a hardness of 372 - 437 BHN. The K-27 ring also is a centrifugally cast martensitic nodular cast iron with a hardness of 250 - 283 BHN. The K-6E ring is a statically cast gray iron (3.7% C - 2.85% Si - 0.45% Mo - 0.3% Cr) consisting of free graphite flake in a fine pearlite matrix and has a

Table 3-2

Cylinder Liner/Piston Ring Material Combinations for Heavy Duty Diesel Engines

Cylinder Liner	Piston Rings Type	Piston Rings Material
1. Hard Chromium Plated Centrifugally Cast Gray Cast Iron	Top Compression	K-28 Nodular Cast Iron
	Bottom Seal	K-6E Gray Cast Iron
	Oil	K-Iron
2. Nitrided Centrifugally Cast Gray Iron	Top Compression	Hard Chromium Plated
	Bottom Seal	K-Iron
	Oil	K-Iron

hardness of 230 - 297 BHN. The K-iron (3.7% C - 2.65% Si) is a statically cast unalloyed gray cast iron consisting of free graphite flake in a matrix of peralite and with an ultimate strength of 30,000 psi.

3.1.3 Piston

The pistons for automotive internal combustion engines are produced primarily from aluminum die coatings. The light weight and excellent thermal conductivity of aluminum alloys made them very attractive as the trend in engines went toward higher speeds and higher compression ratios. Most pistons are made from SAE 328 (Al, 11 - 12.5% Si, 0.9% Fe, 1 - 2% Cu; 0.5 - 0.9% Mn, 0.4 - 1.0% Mg, 0.05% Ni, 1.0% Zn, 0.25% Ti) and SAE 332 (Al, 8.5 - 10.5% Si, 1.2% Fe, 2 - 4% Cu, 0.5% Mn, 0.5 - 1.5% Mg, 0.5% Ni, 1.0% Zn, 0.25% Ti). These alloys are usually heat treated (T5) to a hardness of approximately 100 BHN and are tin plated for scuff resistance.

For heavy duty engines, the piston head usually is produced from alloyed (Cr-Mo) cast iron. However, in order to achieve greater reliability, at least one manufacturer has changed to a cast Mn-Mo steel (0.20% C - 1.35% Mn - 0.5% Si - 0.25% Mo) in the production of power piston heads. The Mn-Mo steel has a minimum yield strength of 60,000 psi and a hardness of 180 BHN. In order to reduce weight of the cast steel power piston, the lower portion is constructed from a forged aluminum alloy - usually 4032 alloy (11 - 13.5% Si, 0.8 - 1.3% Mg, 0.5 - 1.3% Cu, 0.5 - 1.3% Ni) in the T-6 condition. The 4032-T6 alloy has a minimum yield strength of 42,000 psi and a hardness of 115 BHN.

One of the major reasons that the high silicon content aluminum alloys are used in the fabrication of aluminum pistons is to minimize the difference in thermal expansion between the cylinder wall material and the piston material. The coefficient of thermal expansion for 4032 alloy between RT and 572°F is reduced to 11.7×10^{-6} in./in./°F from 13.1×10^{-6} in./in./°F for alloyed aluminum.

3.1.4 Piston Pin

Failure of piston pins rarely occurs. However, 0.001 - 0.002 inch wear of the piston pin results in a noisy engine so that selection of materials to minimize wear is very important. Piston pins are normally produced from AISI 1117 (1.15% Mn), 1016, 5115 or 5120 (0.8% Cr) steel for automotive engines and AISI 8620, 8640 (0.55% Ni - 0.5% Cr - 0.2% Mo) and 5046 (0.4% Cr) alloy steel for heavy duty engines. The low carbon steels are carburized and the higher carbon steels are case hardened by induction to a surface hardness of Rockwell C 60. Usually the surface is polished to a high finish, i.e., on the order of 2 RMS. It has been determined experimentally that the load that the piston pin can withstand without scoring is directly proportional to the surface finish - the better the finish the higher the permissible load. Under severe conditions of loading, the surface of the pin can be roughened by shot peening (followed by lapping to remove the displaced metal) to form depressions in the surface in order to better hold the lubricant.

3.1.5 Piston Pin Bearing

There are a relatively large number of metals and metal alloys that are used in sleeve bearings. The bearing alloys can be grouped in the following classes: tin-base alloys, lead-base alloys, copper-lead alloys, tin-bronze, silver, aluminum alloys, zinc-base alloys, gray cast irons, non-metallic materials (PTFE) and overlays. The proper selection of the bearing material will depend upon the type of load, i.e., steady state or cyclic, amount of load, speed, temperature, corrosive conditions, oil supply, dirt contamination, and shaft hardness.

Type of load - the load carrying capacity of cyclically loaded bearings, as in piston pin bearings, is usually determined by the bearing fatigue strength of the material. This property is of minor importance for bearings under constant load.

Amount of load - cyclic loads up to 1500 psi are considered low and can be supported by ordinary babbitt bearings (tin or lead base alloys). Thin babbitt overlays can operate up to 2500 psi. The limit of alternating loads for the copper-lead bearing material is approximately 4000 psi. A load of 5000 psi can be supported by a trilayer bearing consisting of a mild steel (AISI 1010) back, a 25% Pb-Cu alloy intermediate layer and a plated babbitt overlay. The latter bearing is used in some heavy duty engines.

Speed - babbitt alloys are best suited for high speed operation.

Temperature - strength and corrosion rates are affected by the service temperature. Certain excellant bearing materials, such as babbitts, are restricted by their low melting points. The tin-bronzes such as SAE 62 (86.7% Cu - 10% Sn - 10.3% Pb - 2.0% Zn - 1.0% Ni) and phosphor bronzes such as SAE 64 (78% Cu - 10% Sn - 10% Pb - 0.75% Zn - 0.5% Ni - 0.5% Sb) can be used as bearings at temperatures approaching 600°F. Maximum loads for these materials is $\sim$ 4000 psi. Both materials have excellant fatigue properties but poor anti-seizure, conformability and embeddability properties; anti-seizure can be improved by silver plating.

Corrosion Resistance - the tin-base babbitt alloys are the most resistant to the corrosive action of acids that are formed in lubricating oils.

Oil Supply - a common cause of failure in bearings is the loss of the oil supply or breakdown in the oil film. This results in direct contact between the pin and the bearing surfaces and a large increase in friciton. When this happens, the anti-seizure characteristics of the bearing are most important. The rating of materials for this property in order of decreasing anti-seizure qualities are: tin babbitt, lead babbitt, aluminum alloys, copper-lead alloys, leaded bronzes, silver and bronze.

Dirt Contamination - in bearing applications where there is a high level of contamination by particulate matter, the bearing material must possess good embeddability. Dirt can be embedded in soft babbitt materials without doing harm.

Shaft Hardness - the harder the bearing, the harder the shaft must be to prevent damage to the shaft. In general applications, soft babbitt bearings can be used satisfactorily with steels as soft as 130 - 165 BHN; for copper lead (to 25% Pb) the minimum hardness of the steel shaft must be 165 - 200 BHN; for aluminum alloys the minimum hardness of the steel shaft must be 200 - 300 BHN; and for leaded bronze the minimum hardness of the steel shaft must be 300 - 400 BHN.

A listing of recommended bearing materials for various bearing loads in internal combustion engines is given in Table 3-3. The materials that are currently being used for piston pin bearings in the two leading large diesel engines for locomotives are (1) a leaded tin bronze (80% Cu - 10% Sn - 10% Pb), both as a solid bushing and with a AISI 1010 steel backing (SAE 792) and (2) a tri-metal bearing consisting of a AISI 1010 steel backing, a 0.015 in. thick silver intermediate layer and a 0.0003 in. thick lead overlay. These bearings require a good finish and many axial grooves for pressurized oil lubrication. A crankshaft bearing material that is currently in use in automotive engines is a 0.030 in. thick SAE 780 (8280) aluminum alloy sheet (6% Sn - 1% Cu - 0.5% Ni - 1.5% Si) bonded to a 0.001 in. thick Ni plated AISI 1010 steel backing. A recommendation received from one of the leading aluminum producers also called for the use of a leaded tin bronze bearing for the piston pin bearing and a AISI 1010 steel backed aluminum alloy bearing for the crankshaft.

3.1.6 Connecting Rod

The high alternating loads imposed on the connecting rod in reciprocating engines require the use of relatively high strength steels of high quality. For this reason connecting rods are generally produced from contour forgings made from vacuum degassed ingots. The most common steels used for the production of connecting rods are AISI 1041 for use in automotive engines and AISI 8640 (0.55% Ni - 0.5% Cr - 0.2% Mo), 4140 (0.95% Cr - 0.2% Mo) and 4340 (1.8% Ni - 0.8% Cr - 0.25% Mo) for heavy duty and diesel engines. One automotive engine manufacturer utilizes SAE 80002 malleable iron castings for the production of connecting rods.

3.1.7 Camshaft/Cam/Cam Follower (Tappet)

The cyclic loading and high localized surface stresses that occur as a result of the action between the cam surface and cam follower (tappet) makes the materials selection for these components very important. In some engines, the localized Hertzian stress on the cam surface approaches 200,000 psi. Numerous combinations of camshaft/tappet materials are in common use. Hardenable gray cast iron, such as SAE G4000 is most widely used for automotive camshafts. A typical composition is 3.3% C - 2.25% Si - 0.6% Mn - 0.95% Cr - 0.5% Mo. As cast, the camshaft has a hardness of

Table 3-3

Bearing Materials Used in Internal Combustion Engines

Bearing Load, psi	Lowest Cost Material	Alternate Materials	Advantages of Alternate Materials
	(Small and Medium Size Engines)		
1200 max	Pb Babbitt on steel (0.015–0.030 in.)	Zr or cast Al alloy	(a) (b) (c)
		Sn Babbitt on steel (0.015-0.030 in.)	(d)
1200 - 2200	Pb Microbabbitt on steel (0.002-0.005 in.)	Babbitt-impregnated sintered Cu-Ni on steel	(a) (b) (d)
		Babbitt-impregnated sintered bronze on steel	(a) (b) (d)
		Sn Microbabbitt on steel	(d)
		Al alloy on steel or solid	(a) (b) (d)
2200 - 2800	Babbitt-impregnated sintered Cu-Ni on steel Babbitt-impregnated sintered bronze on steel Cu-25% Pb on steel Al alloy on steel	Cu-35% Pb on steel	(a) (b)
		Solid Al alloy or Al alloy on steel	(a) (b) (d)
		Above materials + 0.001 in. Pb alloy overlay	(b) (e)
		Cu-25% Pb on steel + 0.001 in. Pb alloy overlay	(d) (e) (f) (g)
		Al alloy on steel + 0.001 in. Pb alloy overlay	(d) (e) (f) (g)
	(Large Engines)		
1200 max	Pb Babbitt on steel or bronze (0.015-0.030 in.)	Sn Babbitt on steel or bronze	(d)
		Solid Al alloy	(c) (d)

Table 3-3 (Cont'd.)

Bearing Materials Used in Internal Combustion Engines

Bearing Load, psi	Lowest Cost Material	Alternate Materials	Advantages of Alternate Materials
		(Large Engines)	
1200 - 1500	Pb Babbitt on steel or bronze (0.020 in. max)	Pb Microbabbitt on steel or bronze	(a) (b)
		Aolid Al alloy	(c) (d)
1500 - 2000	Pb Microbabbitt on steel or bronze (0.002-0.005 in.)	Cu-Pb (30-40% Pb) on steel	(a) (b)
		Solid Al alloy or Al alloy on steel	(c) (d)
		Above materials + 0.001 in. Pb alloy overlay	(b) (e)
2000 min.	Solid Al alloy	Cu-25% Pb on steel + 0.0005-0.002 in. Pb alloy overlay	(a) (b) (c)
		Al alloy on steel + 0.0005-0.002 in. Pb alloy overlay	(d) (g)

(a) Yield strength
(b) Fatigue strength
(c) Possible cost savings
(d) Corrosion resistance
(e) Conformability
(f) Embeddability
(g) Inherent lubricity

248 - 311 BHN; the cam surfaces are flame or induction hardened to a Rockwell C hardness of 54 min. For heavy duty engines the camshafts are forged and are usually produced from water quenched carbon steels and low alloy steels of 0.5 - 0.7% C, AISI 4340, or a carburizing grade such as AISI 8620 steel. Heat treatment of the steel to achieve the necessary hardness of the cam surfaces is accomplished by either conventional through-hardening or case hardening processes using flame or induction. One diesel engine manufacturer uses a forged AISI 1080 shaft which is quenched and tempered to a hardness of 235 BHN. The cam and bearing surfaces are induction case hardened to a hardness of Rockwell C 60 - 65 to a depth of 0.060 in. Care must be taken to insure sufficient depth of case as thin, brittle surface hardened cases tend to spall in service.

Tappets usually fail by scuffing or rapid loss of surface. The most common automotive tappet material is a gray cast iron of a composition similar to 3.2% C - 2.25% Si - 0.8% Mn - 1.1% Cr - 0.6% Mo - 0.55% Ni. The cast iron tappet is hardened by quenching in oil from 1550°F and tempered to a hardness of Rockwell C 55 - 60. Steel tappets also are used and are produced from hardenable steel such as 52100 bearing steel or high-carbon molybdenum steels of the AISI 4000 series or from carburizing grades of steel such as AISI 51200 or 8620.

In some heavy duty engines, where tappet wear has been a severe problem, it was found that the utilization of material containing varying amounts of carbides in the microstructure improved the service life. In one case the successful use of hardened and tempered D-2 tool steel in solving the wear problem was attributed to free carbides in the microstructure. The D-2 alloy has a high carbon content in conjunction with stable carbide formers. The composition is 1.5% C - 12% Cr - 0.8% V - 1.0% Mo. A higher volume carbide content in the microstructure can be obtained with alloys such as Stellite 6B (1.20% C - 30% Cr - 4.5% W - Bal Co) and this material is being used for tappets in some heavy duty engines. In fact, a solid tungsten carbide cermet is being considered for use as a tappet material. In some very heavily loaded cam/tappet designs a roller cam follower is employed. An AISI 8620 steel carburized to a surface hardness of Rockwell C 55 min. and a case depth of 0.060 inch has performed satisfactorily in roller type cam followers. Again a surface finish of 2 - 4 RMS is recommended.

Tappet faces are usually finished to about 6 RMS for the same reasons a good finish is required for the piston pin, i.e., improved load carrying ability. Shot peening of the tappet surface is sometimes done to improve the retention of the lubricant. Coating of the highly finished tappet surface to aid in the wearing-in process and improve frictional characteristics is common practice. An oxide coating (Fe_3O_4) is generally applied to chilled cast iron and a phosphate coating is generally applied to hardened steels or gray cast irons. The phosphates can be manganese phosphate, zinc phosphate or iron-manganese phosphate. Other coatings that have been used are oxidates, manganates, and sulfides.

Bearing materials used for camshafts in heavy duty diesel engines are generally cast aluminum alloys. Aluminum alloy 750 (6.25% Sn - 1.0% Ni - 1.0% Cu) is commonly used.

3.1.8 Inlet Valve

Operating conditions for valves in large central steam power stations vary considerably from the operating conditions of valves for automotive and diesel internal combustion engines. The operating conditions for steam valves are much less severe than for the valves in internal combustion engines. Actuation of steam valves is on an infrequent basis so that fatigue and wear of the valve face and seat is not a severe problem. Service temperatures of the steam valve are significantly lower than those for the exhaust valve of an internal combustion engine and they do not fluctuate as they do in the internal combustion engine; thus creep and and thermal fatigue problems are less severe. In addition, corrosion as a result of high temperature combustion products is not a problem in steam valves. In summary, the years of experience in the operation of steam valves or exhaust valves in internal combustion engines are not directly applicable to the selection of materials for inlet steam valves in reciprocating steam expanders. Selection of materials for the various components of the steam inlet valve will have to be based on the knowledge of the basic properties of the materials with respect to corrosion in steam, mechanical properties and wear behavior under conditions of no lubrication and as supplemented with valve experience in reciprocating internal combustion engines.

Inlet steam valves in large central power steam systems are generally fabricated from a martensitic 12% Cr stainless steel (AISI 410) or Crucible 422 alloy (12% Cr - 1% Mo - 1% W - 0.8% Ni - 0.25% V); the stem and face are nitrided. Both the valve seat and valve guide are made from Stellite 6B. The Stellite 6B on the valve seat usually is applied as a weld overlay on a low alloy steel (2.25% Cr - 1.0% Mo alloy).

The materials in use in valve components for internal combustion engines will vary considerably with the operating temperature of the valve. The lower strength and increased corrosion and wear rates that accompany increased service temperatures necessitates the use of more highly alloyed materials as the temperature of the valve increases. Since the inlet valves in internal combustion engines operate at temperature under 600°F and exhaust valves operate at temperatures as high as 1550°F, it is obvious that the most severe materials problems are associated with the exhaust valve. The development of improved valve materials over the years has been primarily by empirical methods. This stems from the fact that it is not possible to simultaneously reproduce all of the service conditions of the valve by any other means than in an actual engine.

Materials that are recommended for use in inlet and exhaust valves in various types of engines are listed in Table 3-4. The steels in Group A of Table 3-4 are used for light duty inlet valves that operate at low temperature or for short times. They also are used for stem materials in two-piece valve construction where they can be welded to higher alloy heads and used for heavier duty inlet and exhaust valves.

Steels in Group B through E are especially made for valve applications with the Group B steels being the least expensive and the Group E steels being the most expensive. The sigma phase forming steels in Group D have good hot hardness and superior resistance to wear than the austenitic steels of Group E. However, they are more brittle and have less resistance to creep than the Group E steels. The nickel-base super alloys listed in Group F are only used where valve temperatures are very high because of their high cost.

Short operating lives of valves can usually be traced to permanent dimensional changes that take place in the valve components during service

Table 3-4

Materials Used for Inlet and Exhaust Valves in Internal Combustion Engines

Application	Valve Materials	Approx. Max. Exhaust Valve Temp. °F
A. Intake Valve-Light Duty	Carbon Steel	-
	SAE NV-1 (AISI 1041)	
	(a) SAE NV-2 (AISI 1047)	
	(AISI 1050)	
	Low Alloy Steels	
	SAE NV-4 (AISI 3140)	
	(AISI 4150)	
	SAE NV-6 (AISI 5150)	
	(AISI 6145)	
	SAE NV-5 (AISI 8645)	
B. Intake Valve-Heavy Duty	Martensitic Steels	-
	SAE HNV-2 (Sil F) 4 Si-2.25 Cr (0.4C)	
	4 Si-2.25 Cr 1.5 Ni-0.85 Mo (0.4C)	
C. Intake Valve-Heavy Duty Exhaust Valve-Light Duty	Martensitic Steels	1350
	(b) SAE HNV-3 (Sil 1) 3.25 Si-8.5 Cr (0.45C)	
	2.75 Si-7.5 Cr-1.5 Ni-0.85 Mo (0.3C)	
	(b) SAE HNV-6 (XB) 2.25 Si-20.0 Cr-1.5 Ni (0.8C)	
D. Exhaust Valve-Heavy Duty	Austenitic-Sigma Phase Alloys	1550
	SAE EV-1 (SCR) 23.5 Cr-4.75 Ni-2.75 Mo (0.45C)	
	SAE EV-2 (TXCR) 24.0 Cr-3.75 Ni-1.35 Mo-3.75 Mn (0.4C)	

Table 3-4 (Cont'd.)

Materials Used for Inlet and Exhaust Valves in Internal Combustion Engines

Application	Valve Materials		Approx. Max. Exhaust Valve Temp. °F
E. Exhaust Valve-Heavy Duty	**Austenitic Steel Alloys**		1550
	SAE EV-7 (2155N)	21 Cr-5 Ni-5.5 Mn (0.2C, 0.25N)	
		21 Cr-4 Ni-7 Mn (0.4C, 0.1N, 0.22P)	
	(a) SAE EV-8 (21-4N)	21 Cr-4 Ni-9 Mn (0.4C, 0.4N)	
	SAE EV-5 (Sil 10)	19 Cr-8 Ni-3 Si (0.4C)	
	SAE EV-4 (21-12N)	21 Cr-12 Ni (0.2C, 0.2N)	
	(Cast)	25 Cr-12 Ni (0.2C)	
	SAE EV-9 (TPA)	14 Cr-14 Ni-2.4 W-0.35 Mo (0.45C)	
	(Cast)	15 Cr-15 Ni-3.5 Si-0.4 Mo (1.0C, 0.25 Cu)	
F. Exhaust Valve-Heavy Duty	**Austenitic Ni Alloys**		1650
	SAE HEV-2 (Inconel M)	16 Cr-Bal Ni-3 Ti-0.5 Al (0.03C)	
	SAE HEV-3 (Inconel X-750)	15 Cr-Bal Ni-1 Cb-2.5 Ti-0.9 Al (0.04C)	
	(c) SAE HEV-5 (Nimonic 80A)	20 Cr-Bal Ni-2.5 Ti-1.2 Al (0.05C)	
	SAE HEV-6 (Nimonic 90)	20 Cr-Bal Ni-18 Co-2.5 Ti-2.2 Al (0.05C)	

(a) Used in current automotive engines

(b) Used in current high performance automotive engines - intake valves

(c) Used in current heavy duty engines

as a result of inadequate creep strength or poor wear and/or hot corrosion characteristics. The terms used to describe the dimensional changes are: elongation, projection of the stem, face runout and face and tip recession. Elongation ("wire drawing") is the permanent increase in length as measured from the valve face to the tip. This causes a decrease in the "lash" (clearance between valve tip and tappet) so that the valve face does not seat properly and results in "blowby" of the exhaust gases. This can be corrected by using a material with greater resistance to creep. Face runout is caused by hot corrosion which also allows leakage of the exhaust gases. Stem projection is similar to elongation in that it reduces "lash"; it is the total elongation as measured through the valve guide and includes any wear of the seat and valve face and elongation of the stem. Tip recession results in an increase "lash" and leads to fracture of the valve. Face runout and stem projection, due to corrosion and wear of the seat and valve face and tip recession, can usually be corrected by appropriate use of valve seat inserts and/or corrosion resistance and hard facing materials. Commonly used insert and facing materials are listed in Table 3-5. The Group D and E materials are used most often with the cobalt base alloys, Group E, are preferred for improved resistance to corrosion.

Examples of material combinations used in exhaust valve components for current automotive and large diesel internal combustion engines are given in Table 3-6.

3.2 Expander Materials Selection Study

The structural materials used in the containment of high temperature (1000° - 1050°F) and high pressure (2400 psi) steam as well as materials used for specialized components such as steam valves have been studied in depth for years and their properties are well documented. Similarily, materials in use in heavy duty and high performance reciprocating engine components are well established and most of these materials are adaptable to use in corresponding components of the reciprocating steam expander. For this reason, there was no need to conduct experimental investigations of expander materials in Phase I of the program.

Table 3-5

Materials Used for Valve Inserts and Hard Facing Applications

	Material	Nominal Composition, %								SAE Alloy Designation
		C	Mn	Si	Cr	Ni	Mo	W	Co	
A.	Cr-Mo and Cr-Mo-W Steels	0.65	0.6	1.00	3.0	-	5.00	-	-	
	(Inserts)	(a) 1.00	0.6	2.50	4.0	-	8.50	-	-	
		1.00	0.6	0.25	5.0	-	1.25	-	0.25(V)	
		0.65	0.35	0.25	5.0	-	3.00	3.5	1.00(V)	
		1.35	0.35	0.45	3.5	1.0	6.50	5.5	0.20(Cu)	
B.	Cr-Mo Cast Iron	2.50	0.60	2.00	3.0	-	5.00	-	-	
	(Inserts)	2.25	1.00	1.00	2.5	-	7.50	-	-	
C.	W Steels	0.50	-	4.00	-	-	-	9.5	-	
	(Inserts)	0.55	-	0.25	3.5	-	-	13.0	-	HNV-7
D.	Ni Alloys	2.00	0.40	0.30	25.0	60.0	-	8.5	-	VF-4 (X-782)
	(Facings)	0.20	1.00	1.00	19.5	78.0	-	-	-	VF-1 (80-20 NiCr)
E.	Co-Cr-W-Ni Alloys	1.00	-	-	28.0	-	-	4.0	67.0	
	(Facings)	(b) 1.25	-	-	28.0	3.0	-	4.5	55.0	VF-2 (Stellite 6)
		1.60	-	-	24.0	24.0	-	12.5	37.0	VF-5 (Stellite F)
		2.50	-	-	30.0	-	-	11.0	52.0	
		(c) 2.40	-	-	29.0	39.0	-	15.0	10.0	VF-3 (Eatonite)

(a) Light duty on large gasoline and diesel engines.

(b) Strongest and toughest of the Co base alloys.

(c) High hot hardness and resistance to shock and pitting.

Table 3-6

Material Combinations Used in Exhaust Valve Components for Internal Combustion Engines

Exhaust Valve Component	Material(a)		
	Light Duty Spark Ignition	Heavy Duty Spark Ignition	Heavy Duty Diesel
Head	SAE EV-8 (21-4N(b)	SAE HEV-5 (Nimonic) 80A(b)	Inconel 751(c)
Face (overlay)	SAE VF-5 (Stellite F)	SAE VF-3 (Eatonite)	SAE VF-2 (Stellite 6)
Stem	SAE EV-8 (21-4N)(b)	SAE HEV-5 (Nimonic) 80A(b)	AISI 3140(c)
Seat (Insert)	(Eatonite or L.E. Jones Alloy)		AISI 410 SS (Re 40-49) Weld overlay
Guide	←(Gray Cast Iron)→		Gray Cast Iron (Type A Flake in Pearlite)

(a) Material compositions are listed in Tables 3-4 and 3-5.

(b) One piece construction.

(c) Two piece construction - flash welded.

However, prior to the selection of the structural materials for the two single cylinder steam expanders, their strength properties and compatibility in a steam-air environment were reviewed. In addition, material and process specifications were established for each component in the steam expanders.

3.2.1 Material Properties

3.2.1.1 Low Carbon and Low Alloy Steels

The use of any one steel for structural applications and containment of high pressure steam primarily depends on the sustained metal temperatures. Low carbon steel is used for steam generation tubes and the low temperature regions of the superheater where the metal temperatures do not exceed 750° - 800°F. As the temperatures in the steam generating system increases, steels of higher alloy content are employed. The C-0.5% Mo steel can be used up to about 850°F; the low chromium-molybdenum steels (1.0 - 2.25% Cr) up to 1000° - 1050°F; the intermediate chromium-molybdenum steels (3 - 9% Cr) up to 1100°F; and the austenitic stainless steels above 1100°F.

The temperature limits imposed on the various steels are the result of loss in elevated temperature strehgth (yield and creep), structural changes in the microstructure of the steel that may detrimentally affect the properties, i.e., ductility, and oxidation-steam corrosion considerations. The maximum allowable design stresses for the low carbon, low alloy and austenitic stainless steels used in boiler construction as a function of temperature are shown in Figure 3-1. The maximum allowable stress is based on the ASME Boiler and Pressure Vessel Code which limits the allowable stress to the level at which the creep rate is 0.01% in 1000 hours at the design metal temperature. Additional restrictions on the allowable stresses are imposed by the code and they are based on the stress-rupture strength of the material as modified by suitable allowances for loss in metal cross-section due to oxidation and corrosion. From Figure 3-1, the allowable stress for low carbon steels begins to drop above 700°F and at 850°F is only half the value at room temperature. The maximum allowable stresses for the low alloy Cr-Mo steels are sustained up to about 800°F whereupon they start to drop and reach 50% of the room

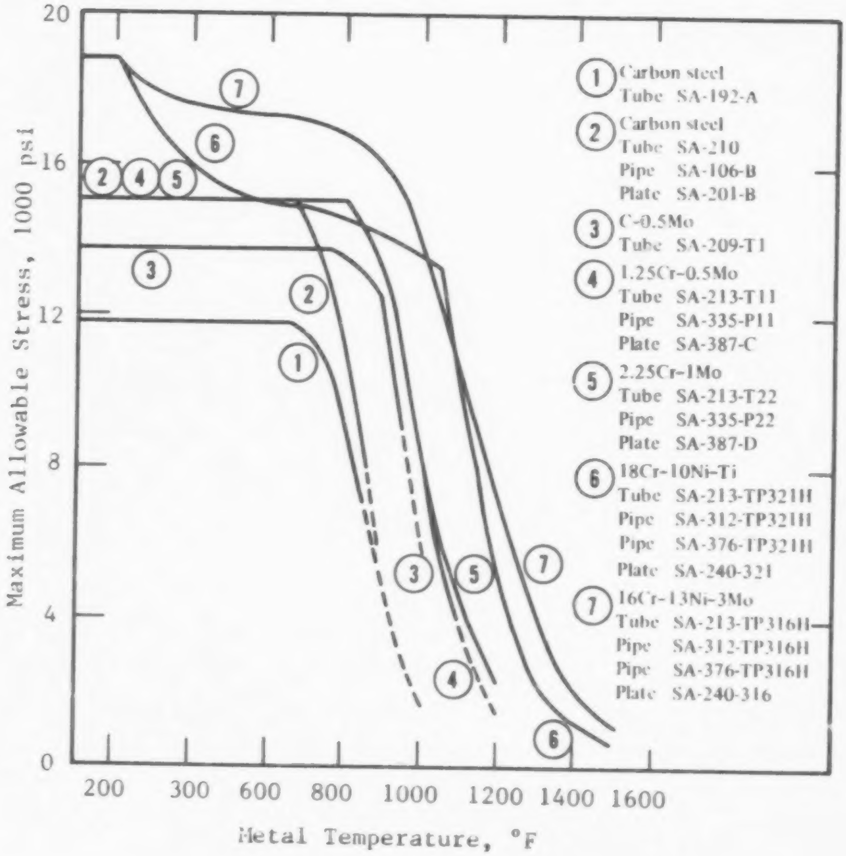


Figure 3-1. Maximum Allowable Design Stresses for Low Carbon, Low Alloy and Austenitic Stainless Steels in Large Steam Generating Systems.

temperature values at 1000°F. The 50% loss in the maximum allowable design stresses for the austenitic stainless steels occurs at about 1200°F. Of the low alloy, Cr-Mo steels, the 1.25% Cr-0.5% Mo alloy has superior creep rupture properties at temperatures on the order of 1000°F, Table 3-7. As chromium is added to achieve superior oxidation resistance, the creep-rupture strengths are reduced. Care must be exercised in the fabrication of components from these steels, i.e., amount of cold work, heat treatment, etc., such that grain size and phase morphology are not adversely affected with respect to their influence on strength and ductility.

The thermal stability of the low alloy steels also restricts the temperature at which the steels can be used. Phase changes, such as the graphitization of carbon steels above 750°F and C-Mo steels above 850°F, can result in drastic reduction in strength and ductility of the steel. Chromium additions of more than 0.5% in the steel eliminate this problem.

Compatibility with the surrounding environment is another important consideration in the selection of the structural material. In the steam expander, as is the case for the boiler, the exterior surfaces of the containment materials are subject to oxidation by oxygen in the air and the internal surfaces are oxidized by the oxygen in the steam. Each steel has a threshold temperature above which rapid oxidation or corrosion takes place generally due to the formation of a thick porous scale. As a result the load carrying ability is reduced and, in the case of boiler tubes, restricted flow and reduced heat transfer efficiency occurs. Sufficient oxidation and corrosion resistance can be achieved in the steels by the addition of small amounts of chromium usually in excess of 1%. The chromium in the steel improves the oxidation resistance by promoting the formation of a tightly adhering scale which inhibites further oxidation. The degree to which chromium additions improve the oxidation resistance of low alloy steels in steam can be seen in Figures 3-2(7) and 3-3(8). In Figure 3-2, the data are plotted as penetration in mils/yr. vs. temperature and in Figure 3-3, the data are plotted as penetration in mils as function of time at 1100°F. It is important to point out that these data (Figure 3-2) were obtained under actual service conditions from samples periodically cut from a large steam plant (Detroit Edison) and show higher corrosion rates than in steady state laboratory tests by a factor of ∿ 2.

Table 3-7

Creep Rupture Properties of Cr-Mo Alloy Steels at 1000°F[a]

Alloy[b]	Stress (ksi) to Produce a Creep Rate of 1% in 10,000 hours	Stress for Rupture, ksi	
		1000 hrs.	10,000 hrs.
1 Cr - 0.5 Mo	11.8	28.4	18.8
1.25 Cr - 0.5 Mo	12.0	27.0	21.0
2.25 Cr - 1.0 Mo	12.0	20.4	15.4
3 Cr - 1.0 Mo	10.5	18.0	14.7
5 Cr - 0.5 Mo	9.0	19.0	14.6
7 Cr - 0.5 Mo	8.0	18.8	14.0
9 Cr - 1.0 Mo	12.0	22.8	19.1

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[a] Reference 10.

[b] Annealed condition.

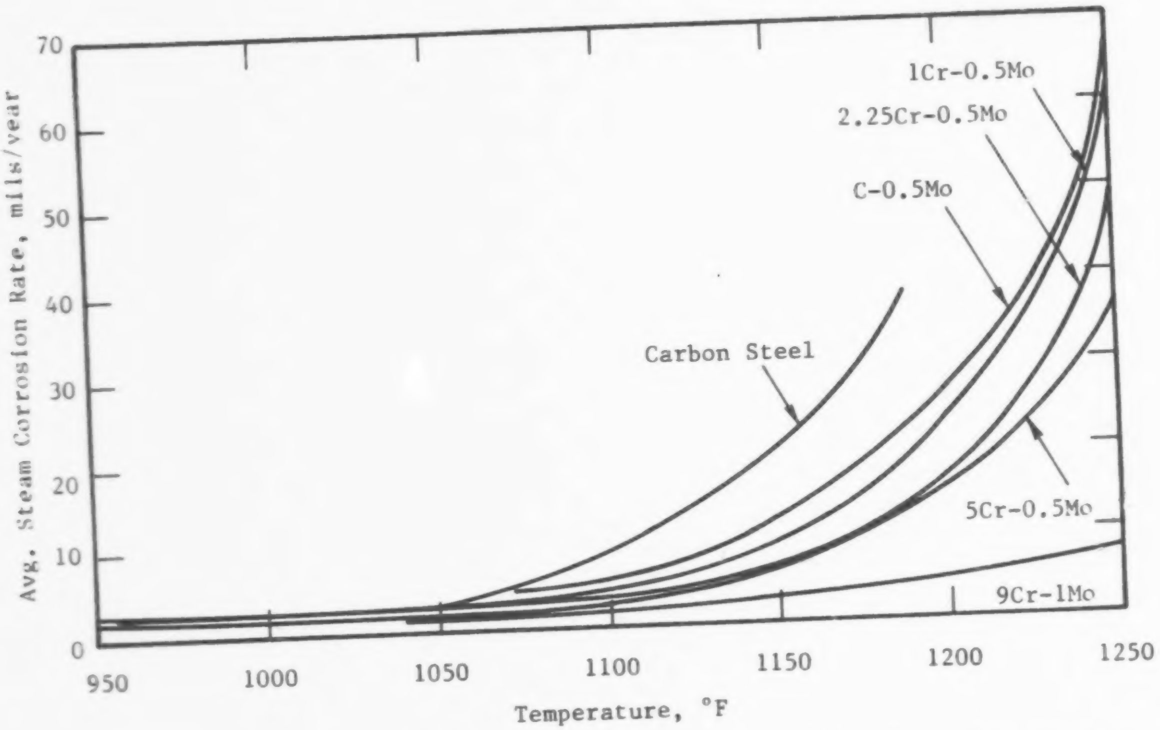

Figure 3-2. Effect of Temperature on Long Time Steam Corrosion Rates.

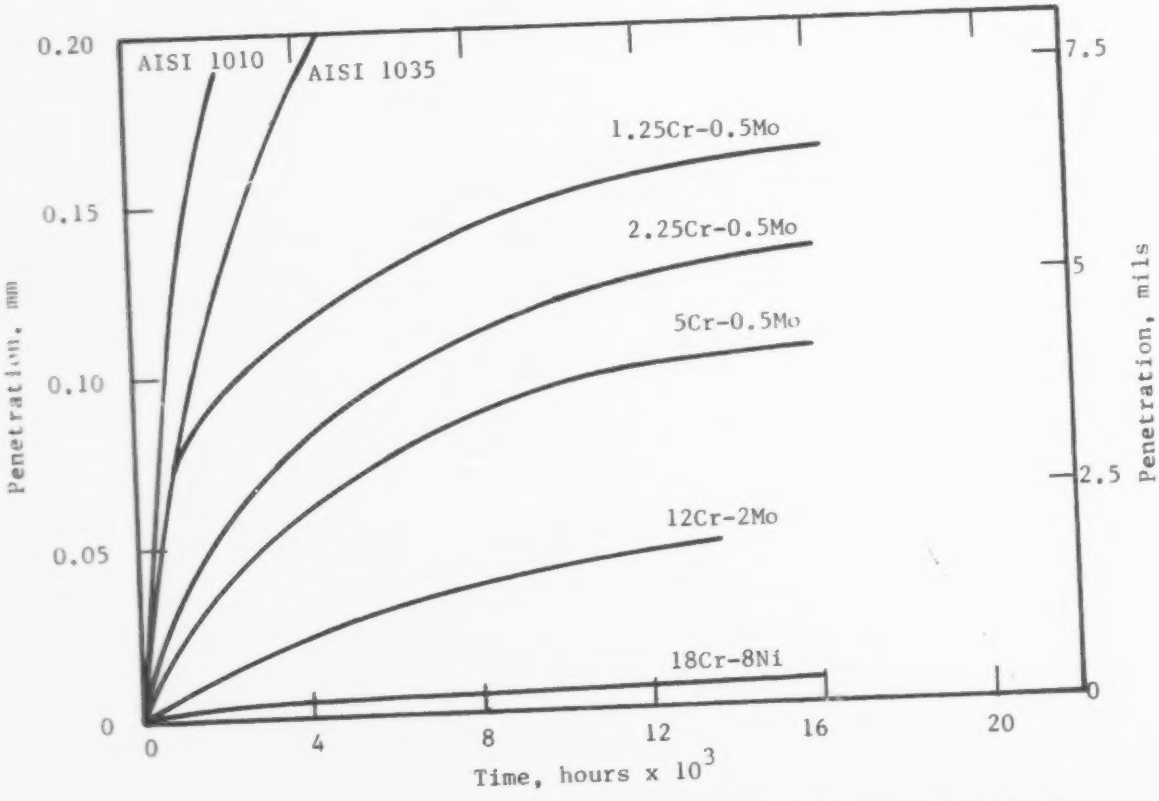

Figure 3-3. Effect of Long Time Exposure to Steam at 1100°F.

Similar long time oxidation-corrosion tests in full size commercial tubing (2 in. OD x 0.5 in. wall) were conducted at the Philip Sporn plant of the American Electric Power Company. In these tests, scaling data were obtained on the internal and external surfaces after varying periods of exposure at 1100°, 1200°, 1350°, 1500°F and 2000 psi steam pressure. Data for the 1100°F exposures are given in Table 3-8. There is little difference in scale thickness between the two surfaces(9).

3.2.1.2 Medium Carbon Low Alloy Steels

A number of dynamic components in the steam expander require the use of high quality relatively high strength materials. For these components, i.e., power piston head, piston rod, connecting rod and bolts, the following medium-carbon, low alloy steels were reviewed:

	Nominal Composition, %				
Alloy	C	Ni	Cr	Mo	V
AISI 4140	0.4	-	1.0	0.2	-
AISI 4340	0.4	1.8	0.3	0.25	-
17-22-A	0.45	-	1.0	0.55	0.3
H-11	0.35	-	5.0	1.5	0.4

The 0.2% yield strength of these alloys are compared in Figure 3-4. H-11 steel has an exceptional combination of high strength and toughness and is used for critical and highly stressed components. When the alternating stress endurance strength is the limiting criteria, vacuum melted grades of the steels are specified. For example, the endurance limits for vacuum melted and air melted AISI 4340 alloy steel heat treated to a strength level of 200,000 psi are 105,000 psi and 90,000 psi respectively at room temperature.

The oxidation steam corrosion characteristics of the medium-carbon, low alloy Cr-Mo-V steels are similar to the low-carbon, Cr-Mo alloy steels. Data for a Cr-Mo-V steel are given in Table 3-8.

3.2.1.3 Cast Iron

For reasons of cost and in certain wear applications, cast iron should be specified wherever possible. Cast iron grades usually are

Table 3-8

Oxidation-Corrosion of Cr-Mo Steels at 1100°F(a)

Alloy(b)	Exposure time, months	Scale Thickness, mils	
		Steam Side	Air Side
Cr-Mo-V(c)	6	← 7.2 total →	
	12	3.3	4.0
	18	4.0	5.0
2.25 Cr - 1.0 Mo	6	2.9	2.9
	12	3.6	3.4
	18	4.1	4.6
	24	5.5	5.7
3 Cr - 1 Mo	6	2.5	2.5
	12	4.8	3.4
	18	4.1	4.2
5 Cr - 0.5 Mo	6	2.8	3.2
	12	3.9	3.5
	18	3.5	2.8
	24	5.5	4.6
9 Cr - 1 Mo	6	2.4	2.1
	12	3.4	3.7
	18	3.8	4.0
	24	5.0	5.4

(a) Steam pressure 2000 psi.

(b) 2 in. OD x 0.5 in. wall tube.

(c) Cr-Mo-V Steel, 0.11 C - 1.0 Cr - 0.86 Mo - 0.22 V.

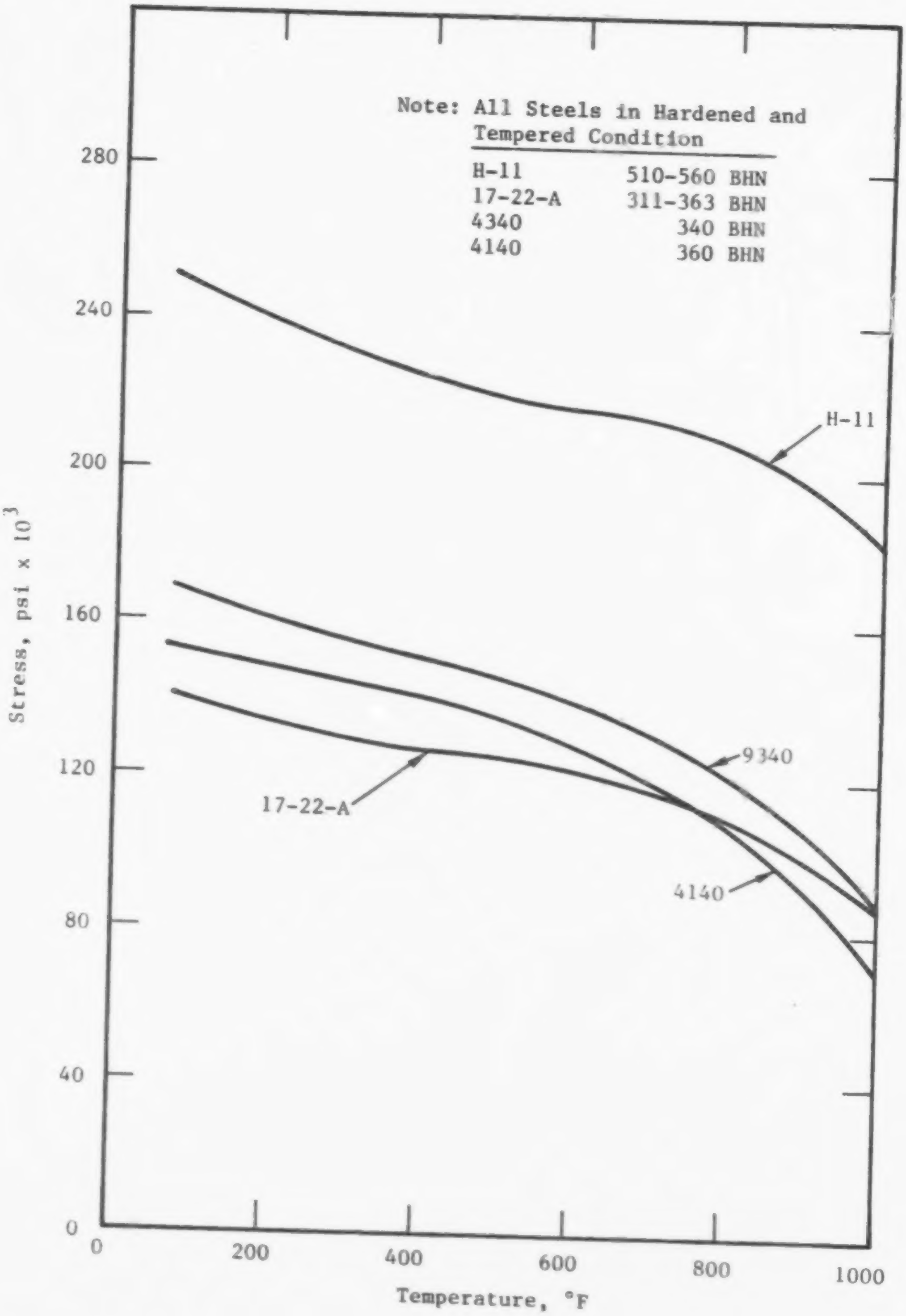

Figure 3-4. 0.2% Yield Strength Medium Carbon Low Alloy Steels.

specified to a specific strength level rather than to a chemical composition, i.e., SAE G4000, and numerous grades of cast iron are available. The ultimate strengtns and endurance limits of gray cast iron generally are not affected by temperature until above 800°F. For example, a 2.84C - 1.5 Si gray cast iron with an ultimate strength of 48,400 psi has a fatigue endurance limit of 20,000 psi. With the exception of a slight dip in strength levels at the intermediate temperatures, to 42,000 psi and 18,000 psi respectively, the initial strength levels are retained to a temperature of approximately 800°F after which they decrease to corresponding levels of 35,000 psi and 14,000 psi. Notched fatigue strength usually are within 10% - 20% of the unnotched strength level.

Although improvements in the high temperature strength of gray cast irons are possible by alloying with molybednum, where reliability is required in critical components of high pressure systems, the use of nodular cast iron is preferred over gray cast iron. Nodular cast iron can be made to significantly higher strength levels than gray cast iron in addition to being able to deform plastically. Measurable tensile elongation values of 3 - 30% are possible with the nodular irons. They also have good resistance to mechanical shock. Typical room temperature properties for a 80-60-03 Pearlitic Grade of nodular iron are as follows:

			Endurnace Limit, ksi(a)	
Ultimate, psi	Yield, psi	Elong., %	Unnotched	Notched
95-130	55-80	3-9	40	24

(a) For casting with 100 psi ultimate strength.

One of the major concerns in the use of cast irons in steam (or air) at elevated temperatures is their dimensional instability due to oxidation. This growth in dimensions can be significantly large in gray cast iron because of the continuous network of graphite. However, additions of chromium and the use of nodular cast iron or, for more severe conditions of corrosion, the use of the Ni-Resist cast irons can eliminate or minimize the dimensional changes due to oxidation. Data on dimensional growth for various cast irons as a result of exposure to steam at 900°F are given in Table 3-9.

Table 3-9

Dimensional Growth of Cast Irons in High Temperature Steam[a]

Cast Iron	Growth at 900°F, in./in.		
	500 Hrs.	1000 Hrs.	2500 Hrs.
Gray Cast Iron	0.0023	0.0052	0.014
Gray Cast Iron + 0.46% Cr	-	-	0.001[b]
Gray Cast Iron + 0.6% Cr	-	-	0.0005[b]
Ni-Resist Type 2[c]	0.0005	0.0010	0.0015
Ni-Resist Type 3[c]	0.0003	0.00045	0.00048
Ni-Resist Type 2D[c] (Nodular)	0.0003	0.0005	0.0005
Ni-Resist Type 3D[c] (Nodular)	0.0003	0.0000	0.0000

---

[a] Reference 11 and Reference 12

[b] 1000°F/2000 hours.

[c] Ni-Resist Type 2, 2D: 0.0C max-2.25 Si-20 Ni-2.1 Cr.
Ni-Resist Type 3, 3D: 2.6C max-2.1 Si-30 Ni-3.0 Cr.

3.2.1.4 Aluminum Alloys

The use of aluminum alloys for dynamic components and cylinder walls in internal combustion engines is highly desirable because of the low density and high thermal conductivity of aluminum alloys. However, the calculated temperatures in the crown of the power piston head, particularly for the crosshead piston expander, and the calculated stresses in the connecting rods and the piston rod of the crosshead piston expander precludes the use of aluminum alloys for these applications. At this stage in the expander designs, only the skirt portion of the trunk piston and the crosshead piston itself warrant consideration. For these applications, it is desirable to utilize alloys with as low a coefficient of thermal expansion as possible that is consistent with suitable mechanical properties.

As previously stated, the following alloys (high silicon contents) are used in piston heads and skirts in automotive spark ignition and large diesel engines: SAE 328, SAE 332 and 4032. The nominal compositions and the coefficients of thermal expansion are given in Table 3-10. Two other high strength forging alloys (2014 and 2219) that have been recommended for high temperature engine components also are listed. For comparative purposes, the room temperature minimum tensile properties of these alloys are given in Table 3-11.

Typical properties of the forged 4032 alloy that was selected for the piston components in this program together with typical properties for 2014 and 2219 alloys are listed in Table 3-12. From these data it is apparent that the 4032 and 2014 alloys have poor thermal stability at the higher temperatures in contrast to the excellant thermal stability of 2219 alloy. Significant degradation in the tensile properties is not observed in the 2219 alloy until a temperature in excess of 600°F is reached or after very long time exposures at 600°F (10,000 hrs.).

Careful consideration must be given to the use of aluminum alloys in steam. Although aluminum alloys should perform satisfactorily in dry steam at elevated temperatures, i.e., 600° - 700°F, erosion may be a problem in wet steam. Further, it is known that conventional commercial aluminum alloys will corrode very rapidly in high purity water at temperatures over 400°F. However, aluminum alloys have been used as cladding

Table 3-10

Nominal Compositions of Aluminum Alloys

	Nominal Composition, %									
Alloy	Si	Fe	Cu	Mn	Mg	Ni	Zn	Ti	Other	Coef. Expansion in./in./°F x 10^{-6}
SAE 328(a)	11.75	0.9	1.5	0.7	0.7	0.0005	1.0	0.25	-	11.5 (68-392°F)
SAE 332(a)	9.5	1.2	3.0	0.5	1.0	0.5	1.0	0.25	-	12.0 (68-392°F)
4032(b)	12.25	1.0(c)	0.9	-	1.1	0.9	0.25(c)	-	0.1 Cr(c)	11.7 (68-572°F)
2014(b)	0.8	-	4.4	0.8	0.4	-	-	-	-	13.6 (68-572°F)
2219(b)	0.2(c)	0.3(c)	6.3	0.3	0.02(c)	-	0.1(c)	0.06	0.1 V 0.18 Zr	13.6 (68-572°F)

(a) Casting - automotive engines.

(b) Forging - large diesel engines.

(c) Maximum.

Table 3-11

Room Temperature Tensile Properties of High Silicon Aluminum Alloys

Alloy	Condition	Temper	Minimum Tensile Properties		
			Ultimate, psi	0.2% Yield, psi	Elong., %
SAE 328	Cast	T5	32	26	low
		T65	42	37	low
SAE 332	Cast	T5	31	-	low
4032	Forged	T6	52	42	5
2014	Forged	T6	65	55	10

Table 3-12

Typical Mechanical Properties for Aluminum Alloy Forgings (4032, 2014 and 2219 in T6 Condition)

Alloy	Test Temp., °F	Exposure Time Prior to Test, hours	Tensile Properties			Young's Modulus ksi x 10^3	Fatigue Properties	
			Ult., ksi	Yield, ksi	Elong., %(4D)		Cycles	Stress, ksi
4032	75	-	55	46	9	11.3	1×10^8 5×10^8	18.0 16.5
	300	1/2	46	41	9	10.5	1×10^8	13.0
		1000	44	40	9		5×10^8	11.5
	500	1/2	23	22	12	9.3	1×10^8	5.5
		1000	10	7	45		5×10^8	5.0
2014	75	-	70	63	15	10.5	-	-
	300	1/2	51	49	14	9.6	-	-
		1000	34	30	20	9.6	-	-
	500	1/2	25	24	18	8.5	-	-
		1000	11	9.5	43	8.5	-	-
2219	75	-	64	45	10	10.6	1×10^8 5×10^8	16.5 15.0
	300	1/2	51	39	20	9.9	-	-
		1000	49	37	20	9.0	-	-
	500	1/2	30	22	25			
		1000	30	22	25	8.6	-	-
	600	1/2	22	16	26			
		1000	18	14	28	7.6	-	-
	700	1/2	12	10	30			
		1000	5	4.5	80	6.4	-	-

materials in pressurized water reactors operating in excess of 300°F. In fact, a new Al-Ni-Fe alloy (X8001) has been developed specifically for this purpose and has shown outstanding resistance to corrosion in high purity water at elevated temperatures. Corrosion tests on the X8001 alloy containing 0.5% Fe and 1.0% Ni in high purity water exhibited the following corrosion rates:

Exposure[a] Temperature, °F	Corrosion Rate, mils/yr
550	1.7
600	3.1
680	8.0

---

[a] Test duration - approximately 1500 hours.

3.2.1.5 Case Hardened Low Alloy Steels

Dynamic components that are relatively low stressed but are subject to wear in service require that their surfaces be hardened. An excellant example is the camshaft where cyclic Hertzian stresses on the surface approach 200,000 psi. In these applications, induction hardening of high carbon steels (AISI 1080), carburizing or nitriding of low-medium carbon low alloy steels (AISI 8620, 4340) are employed. Since fatigue fractures generally initiate at the surface, it is extremely important that the surfaces of case hardened components be free of defects or irregularities that will provide stress-risers. Assuming defect free surfaces, the surface hardening treatments will substantially increase the fatigue endurance limit, partly because of the induced compressive stresses in the surface. As an example, where the fatigue endurance limit of heat treated AISI 4340 is approximately 75,000 - 80,000 psi, shot peening the surface will increase the endurance limit to approximately 90,000 - 100,000 psi and nitriding the surface will increase the endurance limit to 120,000 to 135,000 psi. In a crankshaft application, designed for a minimum 115,000 psi tensile strength, a AISI 4140 steel heat treated to a 85,000 psi tensile strength and nitrided on the wear surfaces, provided the necessary wear and fatigue resistance.

3.2.2 Material and Process Specifications

Material procurement specifications for all the components of the expanders were reviewed and specific specifications were recommended. ASTM, AMS, SAE and General Electric specifications were utilized.

Heat treatment cycles to achieve the desired mechanical properties and dimensional stability were selected for each material and component. In addition, procedures for welding, cleaning, surface hardening were reviewed and specific procedures were established for each component. The following specifications were prepared specifically for this program:

	Specification No.	Category	Title
1.	P4AYA25 (2 classes)	Cleaning	Cleaning of High Carbon and Low Alloy Steel
2.	P4AYA26 (6 classes)	Cleaning	Cleaning of Ferritic, Martensitic and Austenitic Steels and Heat Resistant Alloys
3.	P4AYA27	Cleaning	Cleaning of Low Carbon Steel
4.	P8AYA16	Welding	Welding of Austenitic Stainless Steels to Cr-Mo Alloy Steels
5.	P11BYA12	Surface Treatment	Nitriding of Austenitic Stainless Steels
6.	P11BYA11	Surface Treatment	Carburizing of AISI 8620 Steel
7.	P10CYA10	Heat Treatment	Heat Treatment of AISI 440C
8.	P16DYA10	Plating	Chromium Plating (Electrodeposited)

3.3 Materials Recommendations

The materials recommendations for the single cylinder steam expanders are based on the materials technology survey (Section 3.1) and the materials selection study (Section 3.2). In general, considerable conservatism was exercised in the selection of materials for the engines in this program. The conservatism in materials selection is based on the fact that the engines are to be used as test vehicles. The selection of materials was further compromised by the fact that only one or at most two components of any design are required. For this reason no castings were utilized for major components of the expander other than the valve guide. A listing of the materials selected for each component in the single cylinder steam expanders together with the recommended materials specification and heat treatment condition are given in Table 3-13. These materials were selected on the basis of a high probability of success rather than cost. Future studies can be directed toward materials optimization with respect to cost. In the final selection of materials mechanical properties and environmental compatibility must be balanced against cost. Materials must be used up to the limit of their acceptable life before specifying a more costly material.

Table 3-13

Materials Recommendations for Single Cylinder Steam Expanders

Expander Component	Recommended Material	Material Specification	Condition
1. Inlet Manifold Cylinder Head Valve Body	2.25% Cr-1% Mo Steel	ASTM A387 Grade D, Plate ASTM A335 Grade P22, Pipe	Normalize 1700°F, Temper 1250°F
2. Exhaust Manifold	1.0% Cr-0.5% Mo Steel	ASTM A387 Grade B, Plate ASTM A335 Grade P12, Pipe	Normalize 1700°F, Temper 1250°F
3. Cylinder Block	1.25% Cr-0.5% Mo Steel	ASTM A387 Grade C, Plate ASTM A335 Grade P11, Pipe	Normalize 1700°F, Temper 1250°F
4. Cylinder Block-Lower (Crosshead Piston Expander Only)	AISI 304 SS/Gas Nitride Bore	ASTM A240, Plate ASTM A376, Pipe GE/P11BYA11, Nitride	Anneal 1900°F/Rapid Cool
5. Cylinder Liner (A)	17-22-A/Hard Cr Plate Bore	AMS 6304C/GE B5F5, Forging GE/P16DYA10, Chromium Plate	Normalize 1750°F/1-1/2 Hrs. Mar Quench 650°F/10-15 min. Temper 318 - 363 BHN
(B)	AISI 440C	AMS 5630C, Forging	Austenitize 1900°F/Oil Quench Double Temper 675°F/4 Hrs. Hardness Rc 52-56
6. Piston Compression Rings (A)	Metal Impregnated Carbon-Graphite	See Section 4.0 of Report	
(B)	Koppers K-1051/over Inconel X-750		
(Trunk Only) Rider Ring	Metal Impregnated Carbon-Graphite	-	-
Lower Seal Ring	Koppers K-35 Ni-Resist	-	-
Scraper Seal Ring	Koppers K-Iron	-	-
Oil Cutter Ring	Koppers K-6E	-	-

Table 3-13 (Cont'd.)

Materials Recommendations for Single Cylinder Steam Expanders

Expander Component	Recommended Material	Material Specification	Condition
7. Power Piston	17-22-A (1% Cr-0.55% Mo-0.3% V	AMS 6304C/GE B5F5, Forging	Normalize 1750°F/1-1/2 Hrs. Mar Quench 650°F/10-15 min. Temper 311-363 BHN
8. Piston Rod (Crosshead Only)	H-11 Low Alloy Steel	AMS 6487C, Bars & Forgings	Austenitize 1850°F/Air Cool Double Temper 1000°F/2 Hrs. Hardness Rc 52-55
9. Rod Seal (Crosshead Only)	Carbon-Graphite (P5NR)	-	-
10. Crosshead Piston	Al 4032-T6 Alloy	ASTM B247, Forging	Solution Treat 955 ± 15°F/2 Hrs. Water Quench 150-212°F Age 340°F/10 Hrs.
11. Connecting Rod	AISI 4140	AMS 6382G, Forging	Austenitize 1550°F/Oil Quench Double Temper 1000°F/2 Hrs. Hardness 320-360 BHN
12. Piston Pin	AISI 8620/Carburized	AMS 6276C, Bars & Forging GE/P11BYA11, Carburize	Core Hardness Rc 58 min. Core Depth 0.040-0.070 in.
13. Piston Pin Bushing Bushing	80 Cu-10 Sn- 10 Pb/Mild Steel Backing	SAE 792	Stabilize 1000°F/4 Hrs.
14. Camshaft/Cam, Bearing Surfaces	AISI 8620 Carburized	AMS 6276C, Bars & Forgings GE/P11BYA11, Carburize	Core Hardness Rc 58 min. Core Depth 0.040-0.070 in.
15. Tappet/Tappet Insert	Stellite 6B/Phosphate Antichafing Coating	Cabot Specification/ GE B50A156, Bar AMS 2481C Phosphate Coating	Solution Anneal 2250°F/Tapid Air Cool
16. Tappet Housing	Nitralloy 135 Mod/ Gas Nitride Bore	ASTM A355 Class A, Bar Nitride - Drawing Control	Austenitize 1750°F/Oil Quench Temper 1100°F/2 Hrs. Core Hardness 330-370 BHN Case Hardness Rc 65 (converted) Case Depth 0.015-0.020 in.

Table 3-13 (Cont'd.)

Materials Recommendations for Single Cylinder Steam Expanders

Expander Component	Recommended Material	Material Specification	Condition
17. Push Rod	AISI 8620/Carburize	AMS 6276C, Bars & Forgings GE/P11BYA11, Carburize	Core Hardness Rc 58 min. Core Depth 0.040-0.070 in.
18. Valve Head/Facing	H-11 Steel/Stellite 6	AMS 6487C, Bars & Forgings ASTM A399 Type R Co-Cr-A Hard Surface Rod	Austenitize 1850°F/Air Cool Double Temper 1000°F/2 Hrs. Hardness Rc 52-55 Coating Thickness - 0.125 in.
Valve Stem	H-11 Steel/Gas Nitride	Nitride - Drawing Control	Case Hardness Rc 65 (Converted) Core Depth 0.010-0.015 in.
19. Valve Seat (Facing)	Stellite 3	ASTM A399 Type R Co-Cr-C Hard Facing Rod	Coating Depth 0.125 in. Coating Hardness Rc 48 min.
20. Valve Guide	Ni-Resist D-3 (Nodular)	ASTM A439, Ductile Iron Casting	Stabilize 1600°F/2 hrs. Furnace Cool to 1000°F @ 100°F/Hr. Hold 1000°F/1 Hr. Slow Cool
21. Valve Rings	H.S. 25/LPA-100 Coat	AMS-5759 Coating - Drawing Control	Cold Worked & Aged @ 1100°F/ 5 Hr. Hardness Rc 45-50 Coating Thickness 0.003 in.
22. High Temp. Static Seals	H.S. 25	AMS-5759	Material Cold Reduced to Rc 45-50

4.0 TASK III LUBRICATION

4.1 Lubrication Technology Base - Solid Lubricants

One of the most critical problems associated with the design of an efficient steam engine is that of lubrication within the cylinders and the inlet steam valve. In order to fully utilize the potential thermal efficiency of the steam cycle, it is necessary to operate the expander with high inlet steam temperature. With increased steam temperature, there is a corresponding increase in temperature of the materials within the cylinders and valves. The vast majority of steam expander designs, from the early Stanleys to the most modern engines, obtain upper cylinder lubrication by means of hydrocarbon lubricants pumped in with the steam. Due to thermal degradation, oxidation, and sludging of these lubricants, their use generally limits steam temperatures to about 750°F. In recognition of this limitation, the General Motors SE-101 engine was limited to a maximum steam temperature of 700°F.(13)

The Pritchard engine(14) is lubricated in a similar way. A slow speed mechanical pump forces small quantities of oil into the steam to act as an upper cylinder lubricant. Approximately one pint per 500 miles is used in this fashion. After 10,000 miles of operation with a closed steam cycle, 20 pints (2.5 gallons) of oil will have been pumped into the steam system. A number of other more recent steam engine designs also have expanders with the lubricant added to the working fluid side of the engine.

Even though lubrication technology for steam engines has been established for large, low-speed engines operating at moderate temperatures and pressures, an adequate lubrication system has not been identified for highly efficient steam engines that operate at high inlet steam temperatures and at high inlet pressures. There are two approaches which appear most feasible for lubrication within the cylinders of high temperature, high pressure steam engines: (1) to inject water-miscible fluids, which have been extensively developed for cutting and grinding applications, into the steam; or (2) to use solid lubricants which have been developed for extreme condition applications. With the injection of a water-miscible fluid, it would be necessary to add a filtering system to separate the lubricant from the water to avoid fouling of the steam generator tubes; also, this approach would result in a high consumption of water-miscible fluid. For this reason, use of solid lubricants appears to be the most feasible method of lubricating the cylinder walls in order to permit the utilization of higher steam temperatures.

A literature search and industrial survey was made to identify and evaluate the current technology in solid lubricants and wear resistant materials. Searches were made through the following resources: General Electric Company Automatic Retrieval System, SAE Transactions, Engineering Index, NASA Scientific and Technical Information, AGARD and the Defense Documentation Center. The industrial survey was made to determine those solid lubricants that either are in commercial production or in the stage of advanced development.

4.1.1 Criteria for Solid Lubrication

The most important function of any lubricant is to keep the rubbing surfaces separated, both to prevent wear of the moving parts and to minimize the high coefficients of friction associated with metal-to-metal contact. A solid lubricant keeps the moving surfaces separated solely through its own strength. However, to do so, the solid lubricant must remain in the clearance between the moving surfaces for the entire design life of the engine. It follows then, that in order for a material to be an effective solid lubricant, it must possess most or all of the following properties and characteristics depending upon the form of the lubricant:

1. Good bonding properties or high tendency to adhere to rubbing surfaces
2. Low shear strength
3. High compressive strength

4. Laminar crystal structure
5. High bulk fracture strength
6. Good thermal stability
7. Chemical compatibility with surrounding environment (including structural materials)
8. Low wear
9. Good resistance to thermal shock
10. Good fabricability.

Although all of the above listed properties are important with respect to achieving satisfactory performance by solid lubrication, the ability of the lubricant adhere to the rubbing surfaces is absolutely essential in order to protect the surfaces and prevent scoring, galling, gross welding and resultant metal transfer. This corresponds to wetting in liquid lubricated system. For self-lubricating composites, it is necessary that the lubricant be transferred to the mating surface to form an adherent thin film. For example, the ability to form a transfer film is what makes graphite an excellant lubricant.

Low shear strength is important from the standpoint of achieving a low coefficient of friction. The coefficient of friction for solid lubricants in general is high in comparison to oils, as shown in the following tabulation:

Lubricant	Typical Coefficient of Friction
Oil	0.001
Solid Lubricant-Bonded	0.01
Solid Lubricant-Composite	0.1
Metal-Metal (Steel)	Approaching 1.0

From these data it can be seen that it is desirable to utilize materials with as low a shear strength as possible.

A high compressive strength is desirable from the standpoint that the higher the compressive strength of the lubricant, the higher is the load that can be applied before metal-to-metal contact is made. In effect, the lubricant should have a low shear strength/compressive strength ratio for best performance.

The laminar crystal structure of the lubricant is associated with low shear strengths and low coefficients of friction. Although effective lubrication can be achieved with solids that do not have a laminar structure, the most effective solid lubricants are those that do have the laminar structure and have good surface adherence properties. The laminar structure consists of alternating layers of atoms in which there are strong bonds between atoms within the layer (covalant or ionic forces) and weak bonds between atoms in adjacent layers (Van der Waals forces). The Van der Walls forces are easily broken in the "layer-lattice" structure resulting in the successive atomic layers readily sliding over each other such as basal plane slip in the hexagonal structure of graphite.

Solid lubricants that have a laminar crystal structure are highly anisotropic with respect to mechanical properties. In cases where the lubricant is fabricated into a self-lubricating composite, the lubricant matrix must have sufficient strength to withstand high pressure differentials and frictional drag loads at high temperatures without changing shape or fracturing. To endure the start-up cycle of a steam engine, the solid lubricant material also must have adequate resistance to thermal shock.

Thermal stability of the lubricants is required to assure acceptable performance over the entire temperature range and throughout the design life of the engine. Since the total surface temperature of the lubricant includes the heat of the environment and the heat of friction, the influence of both must be considered. Thermal stability includes resistance to thermal decomposition and crystallographic changes as well as being resistant to oxidation in air and steam. Crystallographic changes will result in changes in frictional characteristics. However, the effects of surface heat is not always detrimental since some solid lubricants are designed to work as a liquid phase.

Another important consideration is compatibility with all engine materials and small quantities of organic lubricants. Consideration of free energy exchange between steam and the candidate materials indicates that a large number of candidate solid lubricants are available for evaluation. However, free energy exchange is somewhat complicated by the addition of cylinder and piston materials.

Finally, an optimum solid lubricant is one that has low wear and the ability to replenish the transfer film such that long life is obtained. Also, for self-lubricating applications, the solid lubricant must be readily fabricable into usable shapes.

4.1.2 Forms and Types of Solid Lubricants

The lubricating solids can be used in several different forms: (a) as loose powder dusted or rubbed on the mating surfaces or supplied to the parts in a controlled stream of a carrier gas; (b) as bonded materials attached to the surfaces with an adhesive binder; (c) as self-lubricating bearing materials where the lubricating solid is dispersed throughout a metal or plastic bearing or where the lubricating solid forms the matrix of a fabricated bearings, and (d) as additives to oils and greases to replace the chemical extreme pressure (EP) additives normally used. The latter application of solid lubricants is not applicable to the lubrication of cylinder walls and will not be discussed further. Of the remaining three forms in which a solid lubricants can be applied, each form has its own advantages and disadvantages.

Loose powders which have the advantage of minimizing fabrication or assembly problems have the disadvantage of providing a very thin film of limited life; the film either has to be replenished by external means in order to achieve the desired life or be used only to assist in the wear-in period.

Bonded solid film lubricants generally provide longer life than powders with the performance of the bonded solid film varying with the specific lubricant, the bonding agent and the method of application. Some dry films depend solely on small particle size for bonding. The lubricant particles are held together only by the attraction arising from the extremely fine particle size and there is little bonding to the substrate. A limited amount of dispersing agent is sufficient to bring about bonding. Particle-bonded dry films can be dispensed from aerosol containers or dispersed in water, organic solvent or other volatile carrier. Most of the particle-bonded lubricants are air-drying and require no baking. However, the particle-bonded lubricants have the same problem as the loose powders in that they have very limited life unless they can be replenished. An organic resin (phenolic, epoxy, polyimide) may be added to increase binding between particles and bonding to the substrate. The resin produces an adherent paint-like film that offers the best wear life and widest range of use. The lubricant particles also can be bonded together by a silicone, a suitable ceramic or metal salt (sodium silicate, aluminum phosphate) that is hardened upon removal of the solvent. This type of film can be very hard and is more temperature-resistant than the organic resin bonded films. The type of binder that is used depends upon the adhesion required, the anticipated service temperature and environment, and the desired wear life. This type of bonded film can be applied by spraying, dipping or brushing. Newer application techniques include electrophoretic deposition, plasma spraying and sputtering.

One of the major problems associated in the bonded films is that the lubricant is usually limited by the binder itself. For example, the resin bonded solid lubricants are limited to use temperatures of up to about 400°F; silicones and silicates have permitted service temperatures to increase to about 600°F at the expense of higher friction and lower wear life. The organic resin and metal salt bonded solid lubricants also have a limited wear life that can be attributed to the very small volume of lubricant available. This becomes evident when one considers that the film is only 0.0002 - 0.0005 in. thick.

Because of the relatively short life of the bonded solid lubricants, self-lubricating components containing solid lubricants have been developed for longer life bearing and seal applications in order to take full advantage of the good lubricating qualities of solid lubricants. The solid lubricants are incorporated into the self-lubricating composites by impregnation, sintering, and various forms of high-temperature, high-pressure compaction. Self-lubricating bearings generally fall into two major classes: (a) solid lubricant matrix utilizing graphite or the lubricant plastics such as nylon, polytetrafluoroethylene (PTFE), etc., with reinforcing materials as fiberglass, graphite, molybdenum disulfide, and soft metals; and (b) metal matrix in which lubricating solids such as graphite, PTFE, metal disulfides, selenides, or tellurides and low melting point metals are mixed with metal/alloy powders of copper, iron, nickel, molybdenum, and etc., and sintered under pressure or impregnated into a porous metal matrix skeleton.

In the application of solid lubrication for cylinder walls in high performance reciprocating steam expanders where steam temperatures approach 1000°F, the peak steam pressure is 1000 psi and peak piston ring side loads are on the order of 200 - 300 psi at an average linear velocity of 1200 -1600 feet/minute, it was assumed that self-lubricating composites would be required to achieve the necessary life. Therefore, in the survey of solid lubricants, emphasis was placed on the self-lubricating solids or composites. Bonded solid film lubricants were considered primarily as an auxiliary lubricant during the wear-in process or as an aid in establishing a transfer film. The following materials were reviewed as lubricants to minimize wear in the cylinder wall/power piston interface and valve stem/guide interface:

Self-Lubricating Solids

- Carbon-graphites
- Polytetrafluoroethylene (PTFE)
- Sulfides/Selenides
- Metals
- Porous metal composites
- Hard surfacing materials

Bonded Solid Films

- Sulfides/Selenides
- PTFE
- Graphite
- Soft Oxides

4.1.2.1 Self-Lubricating Solids

4.1.2.1.1 Carbon-Graphites

Graphite shows promise of being a very effective solid lubricant for steam engine applications. Graphite has been used as a lubricant for many years and is used extensively for brushes in electric motors. In fact, it was excessive wear of motor brushes in high altitude aircraft that led to the detailed study of the mechanism of graphite lubrication. In a detailed study of this problem by the General Electric Company(15,16,17), it was discovered that the low friction and wear normally exhibited by graphite is due not to any lubricant quality inherent in the graphite, but to adsorption on the surface of substances from the atmospheric environment. The laminar structure of graphite undoubtedly is essential, but alone it is not sufficient. It was found that water vapor, adsorbed on the surface, caused reduction in friction and wear rate. The minimum pressure of water vapor that is necessary to achieve minimum wear rates is on the order of 3 - 5 torr. Oxygen showed a similar effect but at much higher pressures, i.e., 300 - 500 torr. It has been proposed that only the edges of the graphite crystals need be covered with the adsorbed film to achieve the low friction and wear rates by permitting the laminar layers of the hexagonal lattice to slip. In the absence of water vapor or oxygen the edges of the laminar layers of atoms are locked by free radicals formed by the evaporation of volatile oxides of carbon. Graphite then is expected to maintain good lubricity in the cylinder wall because of the presence of high pressure steam.

In extremely high temperature applications, graphite tends to oxidize and its uses are restricted to neutral or reducing atmospheres. However, graphite has been successfully used as seals for steam turbine driven electrical generating equipment which tends to verify its stability in high temperature steam. The threshold oxidation temperature of graphite in steam has been listed as about 1300°F(18) and good resistance to oxidation by steam is expected at the maximum proposed cycle temperature of 900°F. This may be further verified by examination of the free energy changes for the two oxidation reactions:

$$C + H_2O = CO + H_2$$

and

$$C + 2H_2O = CO_2 + 2H_2$$

The free energies of these reactions are plotted against temperature in Figure 4-1. As may be seen on this plot, the driving force for the reactions (negative free energy) does not become appreciable until temperatures greater than about 1200°F. This is in essential agreement with the temperature limit cited above. In air, graphite cannot be used

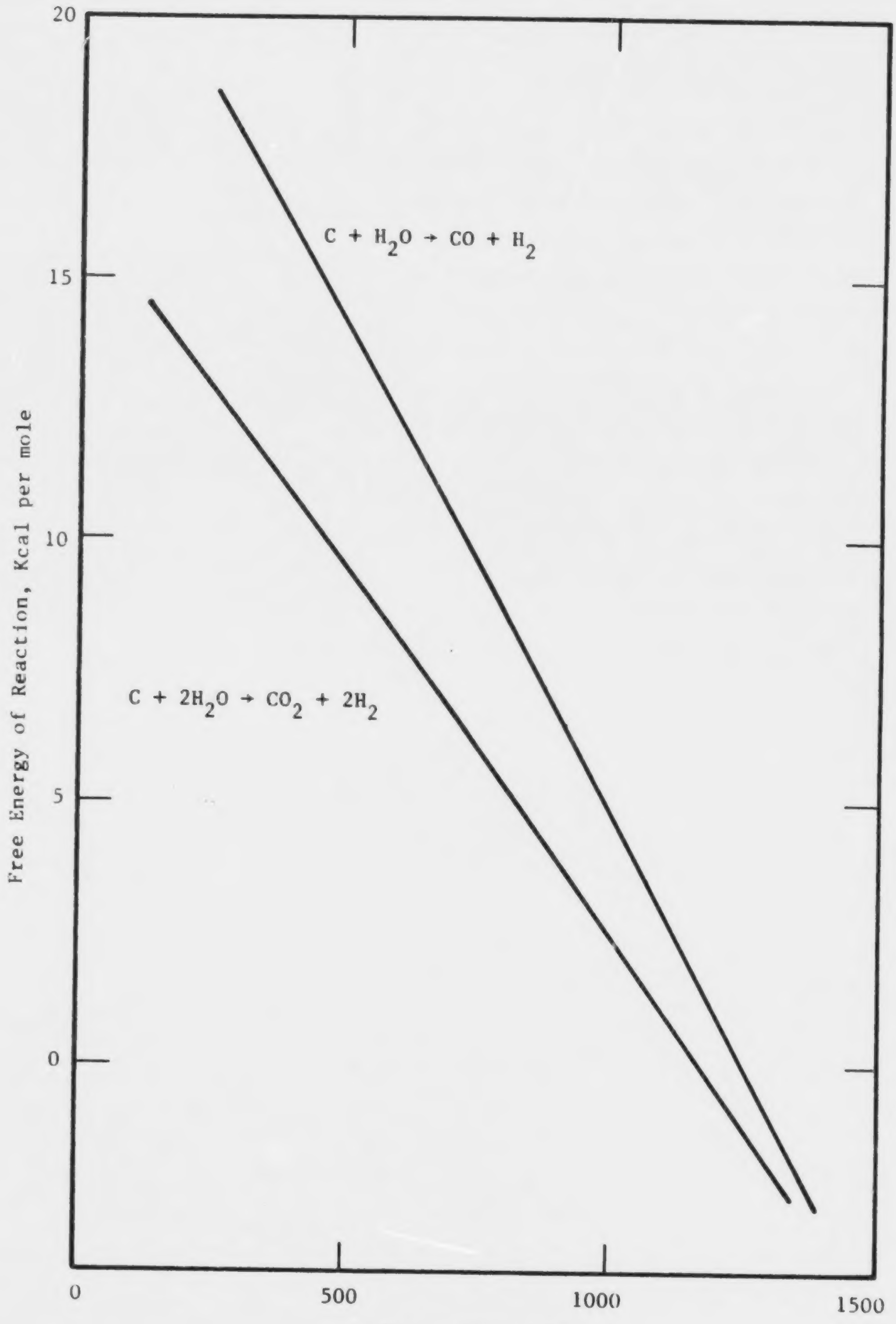

Figure 4-1. Free Energy of Oxidation Reactions Involving Graphite and Steam.

much above 750°F without severe oxidation reactions.

Even though graphite displays excellant properties from the standpoint of thermal stability, thermal conductivity, low elastic modulus, low shear strength and thermal shock resistance, it has poor strength properties in tension. Higher strengths are possible with the carbon-graphite grades of which there are an infinite number to choose; lower wear rates also are possible with the carbon-graphite materials. These materials are available commercially and, although it is an oversimplification, are produced in four general classes: (1) straight carbon-graphites with varying carbon/graphite ratios, (2) resin impregnated, (3) metal impregnated (Cu, Cu-Pb, Cu-Sn, Pb, Ag, Babbitt) and (4) high temperature oxidation inhibited grades. The maximum service temperatures for these materials in air are reported to be up to 500°F for the resin impregnated grades, 700°F for most of the straight carbon-graphites and metal impregnated grades, and 1000° - 1200°F for the high temperature grades. Properties of candidate carbon-graphite grades that are believed to be suitable for application as piston compression rings in a steam environment are listed in Table 4-1. Care must be exercised in the use of the oxidation-inhibited, high temperature grades and some of the resin impregnated grades in steam and water because of the tendency for some of the resins and oxidation inhibitors to exude to varying degrees in these environments. Consideration also must be given to the compatibility of graphite with metals used for the impregnation process as well as mating metals/alloys that are in contact with the graphite in service. Studies by McGee(19) at General Electric has shown that metal oxides that can be reduced to lower oxides or to the metallic state by graphite in the presence of oxygen appear to function as catalysts in the oxidation of the graphite. The relative activity of various metal oxides with respect to their effect on the oxidation of graphite is shown in Table 4-2 and and is based on the change in ignition temperature of pure polycrystalline graphite. The data show that lead oxide is particularily effective in lowering the ignition temperature of graphite.

Although there are numerous influencing factors, the coefficient of friction for the various grades of carbon-graphite will generally range between 0.1 to 0.25. Wear rates also will vary significantly with

Table 4-1

Properties of Carbon-Graphite Grades

Vendor	Grade	Type[a]	Hardness Shore (Scleroscope)	Compr. Str., ksi	Transv. Fracture Str., ksi	Tensile Str., ksi	E 10^6 psi	Coef. Th. Exp., in/in/°F x 10^{-6} (RT-500°F)	Th. Cond. BTU/Ft/Hr °F	Apparant Density gm/cc	Porosity Vol. %	Max. Service Temp., Air
US Graphite Div.	86	CG-R	100	36	12.5	9.5	3.0	2.3	7.6	1.90	0.5	500
	110	CG	92	32	10.0	8.5	3.3	2.3	8.0	1.90	1.0	700
	2980	CG-HT	67	15	4.5	4.0	1.8	2.3	30.0	1.85	4.0	1000
	102	CG-Cu	88	38	9.5	7.5	2.6	3.1	8.7	2.35	4.0	700
	103	CG-Ag	89	35	10.5	7.5	2.8	3.1	9.3	2.70	3.0	700
	107	CG-Sb	92	38	10.0	7.5	3.5	2.4	7.0	2.20	1.0	700
Pure Carbon Co.	P5NR	CG	80	30	8.5	7.0	2.7	2.2	-	1.70	20.0	500
	P03	G	75	20	8.0	5.5	1.7	1.9[b]	-	1.82	10.0	900
	P658RC	CG	90	38	11.0	8.0	3.1	2.2	-	1.80	2.0	500
	X3310	CG-HT	85	29	9.4	8.5	2.6	2.3[c]	-	2.05	0.4	1000
Graphite-Metallizing Corp.	Bronze Graphalloy	CG-Cu-Sn	-	~ 25	-	~ 8.0	-	2.3	-	-	-	750
UCC	CDJN	CG-HT	105	36	8.8	7.0	3.2	2.3	4.4	1.76	-	875
	CJP	CG-HT	65	26	8.0	7.0	2.1	2.3	13.3	1.77	-	1100

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[a]
G Graphite
CG Carbon-Graphite
CG-R Resin Impregnated
CG-HT High temperature/oxidation inhibited
CG-Sb, Cu, Ag, Cu-Sn } Metal Impregnated

[b] RT to 900°F

[c] RT to 1000°F

Table 4-2

Catalytic Activity of Oxides in Graphite Oxidation

Catalyst (acetate or oxide)	w/o as Metal	Ignition Temp., °C
Pb	0.15	384 (738°F)
V	0.20	490
Mn	0.45	523
Co	0.33	525
Cr	0.95	540
Cu	0.20	570
Mo	0.15	572
Ag	0.16	585
Cd	0.21	590
Fe	0.13	593
Pt	0.03	602
Ni	0.45	613
Ir	0.40	638
Rh	0.20	622
Ru	0.30	640
Pd	0.30	659
Ce	0.72	692
Zn	50.00	700
W	0.02	718
Hg	0.10	720
Sn	0.10	738
Uncatalyzed		740 (1364°F)

the grade of carbon-graphite. Overall, the metal impregnated carbon-graphite appears to be the most suitable for use as compression piston rings in the steam expander. Data indicate that after an initial high wear rate during the wear-in period, the metal impregnated carbon-graphites have superior wear rates for long time service; however, the coefficients of friction for the metal impregnated carbon-graphites generally are higher than the other grades. In addition, the metal impregnated carbon-graphites have low porosities and permeabilities, are strong and are compatible with steam. With the exception of their tendency to exude in the pressure of water, some of the oxidation inhibited, high temperature grades also are attractive. They have good friction and wear characteristics and low values of porosity and permeability.

Additional possibilities involve composites of graphite with relatively soft oxides. A cadmium oxide-graphite mixture has friction coefficients in the order of 0.1 or less over most of the temperature range from 100 to 1000°F(20). The cadmium oxide is presumed to improve the adherence of graphite to the surface and, hence, to improve the lubricity effectiveness.

Materials that would be satisfactory as mating materials in contact with the carbon-graphites under dynamic conditions are AISI 440C or hard chromium plate with the latter being limited to service temperatures of less than 700°F. A high hardness is desirable, i.e., greater than Rockwell C 45, and the surface finish should be smooth but not too smooth so as not to be able to hold the transfer film. A finish on the order of 8 - 12 RMS appears satisfactory.

4.1.2.1.2 Polytetrafluoroethylene (PTFE)

Plastic bearings made from polytetrafluoroethylene (PTFE) are available for nearly every type of application. They have been used successfully as inserts to plain bearings, as reinforced thin sheets on plain spherical bearings, and as the retainer material for ball bearings. PTFE is one of those materials that does not have a laminar layer-lattice structure and yet has an exceptionally low coefficient of friction. The low friction has been attributed to its low surface energy. This results in very weak adhesion and shearing takes place primarily at the interface rather than in the bulk material(21). It also has the necessary characteristic of establishing an adherent transfer film on most rubbing metal surfaces. The major disadvantage of PTFE is its low compressive strength which results in cold flowing under relatively low loads and limits its use to temperatures on the order of 400 - 450°F (it also begins to thermally decompose at about 540°F). Other disadvantages of PTFE are its poor thermal conductivity and high coefficient of thermal expansion.

To improve the flow characteristics of PTFE, PTFE composites have been developed. The three most common materials that have been used to reinforce the PTFE are bronze, glass and carbon-graphite. In one recent symposium of PTFE seals for reciprocating air compressors(22), it was generally considered that no one filler material has demonstrated superior performance over the others and that lack of consistency in the composites was a major factor in the erratic performance. Three typical compositions that are in use for seals in reciprocating compressors are:

1. PTFE + (15% Bronze + 5% MoS_2)
2. PTFE + 20% Glass
3. PTFE + 10% Carbon

(% by volume)

G. L. Griffin(23) has suggested that non-uniformities in the structure as a result of molding practice of filled polymers could account for differences in performance. In particular, the surface of molded composites is usually deficient in filler material and this could result in varying friction and wear behavior.

Other developments to improve the poor creep properties of PTFE are the use of fibers of PTFE. Compacts can be made from PTFE fibers which have 25 times the tensile strength of conventional PTFE compacts made from powders.

Halliwell of the U.S. Navy Marine Engineering Laboratory has conducted tests with PTFE reinforced with metallic filament windings(24) in an attempt to develop a piston seal for high-pressure air compressors. The filament winding approach has resulted in a superior reinforced matrix as compared to randomly dispersed particles and fibers previously reported. This new approach to the problem has resulted in extended compressor life with low rates of wear and leakage. Halliwell found that reliable compressor operation could be obtained at 5000 psig for periods beyond 1000 hours using such seals in lieu of conventional split rings. The operational life of PTFE rings in a reciprocating steam expander will be affected by the quality (dryness) of the steam. Wear rates of PTFE decrease with dryness until a dewpoint of ∿ -40°F is reached whereupon the wear rate increases drastically. However, small amounts of condensed water will cause an increase in wear rates and should be avoided.

Polyimides are the next generation of lubricating plastics pushing the use temperature from the 400° - 450°F for PTFE to 500° to 600°F. The polyimides also have superior friction and wear properties. Unfortunately the polyimides cannot be used in high temperature (> 212°F) steam or water because they will hydrolize.

4.1.2.1.3 Metals

In some applications, metals are used on bearing surfaces to achieve improved friction and wear characteristics. The metals (and alloys) used in these applications can be categorized in three classes: (1) classical layer-lattice crystal, (2) low shear strength and (3) soft oxide formers. Buckley and Johnson(25) investigated the influence of crystal structure of metals on their friction are wear behavior and found that relatively low coefficients of friction could be obtained for cobalt in which the hexagonal crystal structure is stabilized. A coefficient of friction of 0.3 was measured for an oxide-free 25% Mo-Co alloy at speeds of 2000 ft/min (750°F; pressure of 10^{-9} torr). The coefficient of friction of cubic cobalt under similar conditions is on the order of 0.7.

A number of metals that do not have the layer lattice structure are still widely used in bearing applications. These metals are listed in Table 4-3; they are soft and shear easily(26). For example, the metals lead, tin, indium and silver are used in many common plain bearings, i.e., piston pin bearing. However, the coefficients of friction of these metals below their melting points are relatively high (> 0.3). When used above their melting points (liquid film) friction coefficients are considerably lower, on the order of 0.17 for lead.

Table 4-3

Soft Metals Used in Bearings

Metal	Moh hardness	Melting point, °C
Indium	1	155
Thallium	1.2	304
Lead	1.5	328
Tin	1.8	232
Cadmium	2	321
Gold	2.5	1063
Silver	2.5-3	961
Platinum	4.3	1755
Rhodium	4.5-5	1955

A commercial material incorporating a soft, low shear strength metal for lubrication is the Bishiralloys(27). These alloys are produced by pressing and sintering alloy powders, followed by impregnation with lead and finally rolling to size. The compositions and hardnesses of these materials are given in Table 4-4. Alloy C is reported to have the best wear resistance; the coefficient of friction is on the order of 0.2. A continuous film of lead is maintained on the surface by flow of the lead in the pores of the alloy. However, because of the low melting point of lead (621°F) the maximum ambient service temperature for the Bisiralloys is about 575°F.

The metal gallium is a liquid at 86°F and effective boundry lubrication is obtained with gallium rich films on AISI 440C stainless steel. The coefficient of friction of gallium coated AISI 440C vs. AISI 440C at 500°F in air was measured to be 0.104 (sliding speed 390 ft/min, load 1000 grams)(28).

Oxidation of metals will result in significant reductions in the coefficients of friction and protection of the metal surfaces(29). The transition temperatures for improved friction are wear behavior were determined for iron, copper, nickel, molybdenum and chromium to be 100° - 200°F, 400° - 500°F, 1200° - 1400°F, 800° - 900°F, 800° - 1100°F respectively. Coefficient of friction values of 0.2 - 0.3 were obtained

Table 4-4

Chemical Composition and Hardness of BISHIRALLOY

	Basic Composition of BISHIRALLOY Matrix Wt. %						wt % Range of Impregnated Lead in BISHIRALLOY	Hardness Brinell
	Fe	Cu	Ni	Mo	Cr	C		
BISHIRALLOY-A	Bal	5.0	--	--	--	less than 0.2	15 ∼ 30	75 - 95
BISHIRALLOY-B	Bal	2.5	1.5	2.5	--	less than 0.2	15 ∼ 30	105 - 135
BISHIRALLOY-C	Bal	2.5	1.0	1.0	5.0	less than 0.2	15 ∼ 30	65 - 95

upon the formation of the oxides in comparison to values of 0.5 - 1.0 for clean surfaces at lower temperatures. The formation of so-called soft oxides MoO_3, WO_3, Cu_2O, ZnO, Co_2O_3, PbO, CdO were the most effective in preventing surface damage. The lowest friction coefficient was obtained with PbO.

A commercial product is available which reportedly provides low friction and wear characteristics as the result of the properties of its oxide surface film. The material is Clevite 300, a iron base alloy containing cobalt and molybdenum. The alloy is produced by powder metallurgy techniques and its properties are listed in Table 4-5.

4.1.2.1.4 Sulfides/Selenides

The sulfides and selenides of molybdenum, tungsten, tantalum and columbium have the layer-lattice structure similar to graphite and have extremely low shear strengths. Further, they do not require water vapor or oxygen for their lubricating ability. Another difference between the sulfides or selenides and graphite is that the weak easily sheared bonds occur only between every third layer of atoms instead of every layer as is the case for graphite. In MoS_2, for example, each layer of molybdenum atoms is strongly bonded to the adjacent layers of sulfur atoms on each side but the layer of sulfur atoms is weakly bonded to the sulfur atoms in the next adjacent layer so that shear occurs through the weak sulfur-sulfur bonds. Recent studies on the lubricating mechanism of MoS_2 were reported at the solid lubrication conference in Denver(23).

Since MoS_2 occurs in nature, it is considerably less expensive than the other sulfides and the selenides or tellirides which are synthetically produced. For this reason most of the common solid lubricant composites based on this group are made from MoS_2. The effectiveness of MoS_2 in air is limited to about 750°F where oxidation begins to effect its lubricity. The oxidation products of MoS_2 are MoO_3 and a sulfur compound which can result in abrasion and corrosion and an increase in the coefficient of friction at the lower temperatures. (It should be noted that at temperatures above 1300°F, MoO_3 can be an effective lubricant; the melting point of MoO_3 is 1364°F).

Table 4-5

Properties of Clevite 300

MECHANICAL PROPERTIES

Temp. (F)	Yield Strength, psi	Ultimate Strength, psi	Elongation, %	Elastic Modulus psi x 10^6	Compression Strength, psi	Hardness RA
75	70,500	106,000	-	28.4	210,000	66 (31 Rc)
600	-	107,000	2.8	25.8	-	57
900	62,250	95,800	2.7	24.4	-	-
1200	48,000	71,000	6.6	22.5	-	35

FRICTION AND WEAR PROPERTIES

(Based on face-seal tests of Clevite 300 against itself at 18 psi load and 150 fps surface speed, in air.)

Temp., °F	Coefficient of Friction, "f"	Wear Rate, in/hr
500	0.072	0.00073
750	0.063	0.00062
1000	0.052	0.00035
1200	0.047	0.00030

Composites based on the solid MoS_2 lubricant have been developed by Boeing(30) and described by Devine(31). The composites developed by Boeing were quite brittle and even the most successful composite, 90% MoS_2 - 8% Fe - 2% Pt, would have strength problems because of the brittle iron sulfides that form at the sintering temperatures. The solid composite called "Navy Lube"(31) is a MoS_2-Graphite-Sodium Silicate ($NaOSiO_2$) and is not only weak but the $NaOSiO_2$ is hygroscopic and would not be suitable for use in the steam environment.

Another series of composites based on MoS_2 and WS_2 are described by Hopkins(32). Three of these composites have shown low wear rates and low friction in air. The compositions of these materials are:

1. 53% WS_2 - 12% Co - 35% Ag
2. 80% MoS_2 - 20% Ta
3. 50% MoS_2 - 38% Ta - 12% Fe

Overall, the use of solid lubricant composites with MoS_2, or other disulfides or diselenides, as the base do not appear attractive for use as piston rings in steam expanders primarily because of their low strengths and fragile nature.

4.1.2.1.5 Porous Metal Composites

In order to utilize the good lubricating qualities of some of the better solid lubricants that are too weak to be used as the base of self-lubricating composites, porous metal composites have been developed. These composites are generally produced by powder metallurgy techniques in which the metal or metal alloy and the lubricant are blended, pressed and sintered to form a metal matrix throughout which are discrete pockets of solid lubricant. In one investigation, it was found that a minimum concentration of 5% of the solid lubricant was required to achieve a transfer film and thus low wear rates. Friction coefficients of 0.2 were measured(33). On the other hand, too high a concentration of the lubricant results in a weak structure. Examples of this type of porous metal composite are: (1) 5-15% MoS_2 in a 95% Ag - 5% Cu matrix, (2) 5-40% MoS_2 in a nickel or 80% Ni - 20% Cr matrix, (3) the commercial Molalloy - MoS_2 in a refractory metal matrix.

Another method of producing the porous metal composites is to prepare a metal skeleton with controlled porosity with respect to volume and distribution and impregnate the pores with the lubricant by application of pressure and heat. Metal skeletons with up to 65% void volume have been produced with Inconel 600, Inconel X-750, Hastelloy X, Nichrome V alloys in which PTFE, graphite, MoS_2 have been impregnated. A Inconel 600 (60 - 65% voids) impregnated with PTFE has performed well in seal applications at temperatures up to 500°F.

The metal fluorides have shown promise for use as lubricants for high temperature application in air. Sliney at NASA Lewis Research Center has investigated the use of fluorides for a number of years(34,35,36). Coefficients of friction of self-lubricating porous metal composites (40% void volume in Inconel X-750), vacuum-impregnated with fused fluoride eutectic of 62% BaF_2 - 38% CaF_2 were measured to be less than 0.1 at 1000°F and at a sliding velocity of 2000 ft/min while under load. The coefficient of friction increases as the temperature and speed decrease; at 500°F and a speed of 1000 ft/min the coefficient of friction is approximately 0.3. More recent studies have shown that coefficients of friction of 0.2 ± 0.05 could be obtained at speeds as low as 500 ft/min over the temperature range of RT to 1700°F. These data were obtained with a porous nickel alloy impregnated with a high temperature enamel (NBS-418) and an overlay of the BaF_2 - CaF_2 eutectic.

Finally, this approach is compatible with classical piston ring materials technology where gray cast irons are used because of the excellent lubricity imparted by flakes of graphite. Similarly, the nodular Ni-Resist alloys of 20 - 30% nickel content with its spheroids of graphite has been shown to have good friction and wear characteristics in conjunction with good corrosion resistance in superheated steam. It is believed that the use of strong porous metal composites impregnated with a solid lubricant of low shear strength and ability to establish a transfer film is one of the most attractive materials for application as piston rings in reciprocating steam expanders.

4.1.2.1.6 Hard Surfacing Materials

In applying the adhesion theory in friction and wear, relatively low coefficients of friction are achieved between two sliding hard materials that have no affinity for each other, i.e., materials with little or no mutual solubility or tendency to form intermetallic compounds. The hard, wear resistant surfacing materials fall into this class of materials and warrant investigation.

Plasma sprayed coatings are of interest because they can be readily applied. Koppers has plasma sprayed rings with their K-1051 coating (refractory metal bonded Cr_3Cr_2) for use in lubricated, reciprocating steam expanders running against nitrided steel or AISI 440C stainless steel which was heat treated to a hardness of Rockwell C 50-55. Since the hard metal carbides are metallic in nature with respect to their atomic bonding, a more suitable materials combination with respect to "alloying tendency" might be an oxide coated cylinder liner running against a carbide coated ring. Possible combinations are Union Carbide Corp. LC-19 (Cr_2O_3 + Al_2O_3) sprayed on the cylinder wall and either LW-1 (Cobalt bonded WC) or LC-1 (Nichrome bonded Cr_3C_2) sprayed on the ring surface. In dry rubbing tests between LC-1 (Cr_3C_2-15% Ni-Cr) and LA-2(Al_2O_3), coefficients of friction of 0.15 - 0.27 were measured in the temperature range of RT to 1400°F.

E.I. du Pont de Nemours and Co. have recently made available a series of anti-friction and anti-wear LP alloys offering outstanding advantages. Metallurgically, the LP Alloys consist of hard grains of an intermetallic compound with a Laves Phase structure dispersed in a softer matrix, that provides good embeddability characteristics in contrast to other hard facing materials. Currently preferred compositions contain Laves-phase hexagonal close-packed intermetallic compounds of cobalt, molybdenum and silicon (Co_3Mo_2Si and CoMoSi) in a cobalt-rich matrix. The Laves phase, depending upon its composition has a micro-hardness of 1000 to 1500 DPH and the matrix has a hardness of 200 to 800 DPH. Overall, the bulk hardness ranges from R_c 30 to R_c 60. Tests have shown that, under extreme loading and boundary lubrication conditions, LP Alloys exhibit very little wear.

LP Alloys are produced as medium and fine powders for plasma spraying and powder metallurgy applications; they also can be cast as semi-finished parts. The compositions (in wt. %) of LP Alloys are shown in Table 4-6.

Table 4-6

Chemical Composition of LP Alloys

	Co	Mo	Ni	Cr	Vol. % Laves
LPA 100	35	35	10	--	65
LPA 200	70	28	2	--	20
LPA 300	45	48	7	--	98
LPA 400	62	23	2	8	50

In one application, the sleeve and roller bearings and the piston skirt and rings of a 2-cycle engine were coated with LPA-100 and the engine operated without oil in the fuel for 50 hours with satisfactory results.

Another new material of interest that is available from E.I. du Pont de Nemours is the compound Ni_3B. The material is applied to the surface by electro-chemical methods and has a microhardness of 900 - 1500 knoop (R_c 60-70). Friction data of Ni_3B against hardened steel show a coefficient of friction of 0.1 at 600°F in air, unlubricated.

Other materials in this class that could be considered are: various grades of SiC, Si_3N_4 and TiC in the form of pressed and sintered compacts or applied as coatings.

4.1.2.1.7 Mixed Composites

Multicomponent self-lubricating composites have been developed that contain a metal matrix, a film former, and a load-carrying component(37). Examples of these types of composites are Ag-PTFE-WSe_2,Cu-PTFE-WSe_2, Ag Bronze-PTFE-WSe_2,Ag Hg-PTFE-$MoSe_2$. The latter material has shown excellant friction and wear characteristics at 600°F(38). The modified PTFE melts at 590°F and is thought to provide the excellant friction and wear characteristics.

Other mixed component self-lubricating composites that have been investigated are: WSe_2-Ga/In(38) and Graphite-WS_2-NaF(39).

4.1.2 Bonded Solid Film Lubricants

The primary disadvantage of the bonded solid film lubricants is that there is little stored lubricant in the film and as a result, service lives are limited. The bonding agents are either organic thermoplastic or thermosetting resins (phenolic, epoxy, polyimide) or inorganic metal salt (sodium silicate) and fused ceramic (aluminum phosphate). Because the bonded films have so little strength, the films are kept as thin as possible, usually on the order of 0.0005 inch or less. For these reasons, the use of bonded films is generally limited to high load/low sliding velocities or low load/moderate velocities. Most commercially available bonded solid film lubricants are limited in temperature by the bonding agent. For example, at temperatures above 400°F, the organic bonding agents tend to thermally decompose causing failure of the film because of lack of adhesion. Commercial products of this type incorporating various solid lubricants are:

Solid Lubricant	Bonding Agent	Service Temp., °F
MoS_2	Epoxy	275 - 300
90% MoS_2-10% Graphite	Phenolic	300
PTFE	Polyimide	450 - 500

A newer bonded film based on the polyimide for use at higher temperatures was reported by Campbell and Hopkins(40). It consists of MoS_2 and Sb_2O_3 dispersed in the polyimide binder (MLR-2) and it shows excellant wear properties. However, it should be pointed out again that the polyimide resins cannot be used in high temperature steam and water.

The metal salt and fused ceramic bonded films were developed for use at temperatures generally above 600°F. Examples of these type of bonded films was reported by Hopkins(41): MLF-5, is a sodium silicate bonded MoS_2-graphite - Au film and MLF-9 is a aluminum phosphate bonded MoS_2-graphite - Bi film. Ceramic bonded MoS_2-graphite films have been used in excess of 700°F in air. As mentioned previously sodium silicate is hygroscopic and cannot be used on steam or water.

At the higher temperatures, > 1000°F, PbO has been shown to exhibit good friction and wear properties. Because PbO oxidizes to Pb_3O_4 at temperatures below 1000°F resulting in loss in lubricity, additions of SiO_2 are made to PbO to inhibit the oxidation reactions(42). Very low friction characteristics (0.1 coefficient) have been achieved with 10% SiO_2-PbO bonded film at temperatures of 500° 0 1200°F.

Another very promising new material that is being studies is graphite fluoride, $(CF_x)_n$(43). Coefficients of friction equal or superior to MoS_2 and graphite were reported at temperatures up to 750°F in air. Superior wear properties are attributed to greater adhesive qualities. Although moisture is benefitical, it is not a prerequisite for low shear strength.

The use of bonded solid film lubricants does not appear to be as promising for the lubrication of the cylinder walls in a steam reciprocating steam expander as do the self-lubricating composites. It would appear that if the bonded solid films were to be evaluated for this application, they should be applied to the cylinder wall rather than the piston rings in order to provide a greater amount of available lubricant for increased life.

4.2 Lubrication Technology Base - Liquid Lubricants

Lubrication of the large, slow speed reciprocating steam expander operating under steam at relatively low temperatures and pressures has been well established for years. On the other hand, lubrication of the small, high speed and high performance steam expander operating at high steam temperatures and pressures is considerably more complex and may be more complicated than the lubrication of internal combustion engines. Using conventional techniques of lubricating steam cylinders and valves by injecting lubricants directly on the parts and/or by injecting the lubricants into the inlet steam, two separate lubricating systems have been used, one for lubricating the components in the crankcase and one for lubricating the cylinder walls and valves that are operating in the high temperature steam. This is accomplished through the use of a cross-head piston design; both the General Motors SE-101 and SE-124 steam engines employed this concept. A further complication is the fact that the cylinder walls cannot be cooled without decreasing the efficiency of the engine cycle so that the fluid lubricating the hot cylinder walls and valves will come in contact with metal surfaces that are at temperatures of 600° - 800°F. Thermal decomposition products of the lubricant resulting from being in contact with metals at these temperatures will tend to foul up the steam generator tubes unless the lubricant is separated from the condensate water. Frequent lubricant changes would be expected.

Successful use of solid lubricants to lubricate the cylinder wall/piston ring and valve stem/valve guide interfaces will eliminate or greatly minimize many of the problems encountered with liquid lubricants. In fact, it may be possible to utilize a standard trunk piston design. However, due to the fact that the engine cylinders cannot be cooled, as stated previously, even the bulk temperature of the crankcase lubricant may be exceptionally high in comparison to the crankcase temperature in conventional internal combustion engines. Crankcase lubricant bulk temperature may be as high as 300°F unless a special cooling system is employed to cool the lubricant. Even so, the limiting temperature may be the hot spot temperature that the lubricant will see on the bottom surface of the power piston head of the trunk piston design or the internal surface of the piston rod in the crosshead piston design. The anticipated bulk lubricant temperature and maximum hot spot temperatures for the two engine designs in this program are:

	Max. Lubricant Temp., °F	
Engine Design	Bulk	Hot Spot
Trunk Piston	250	540
Crosshead Piston	250	425

Although these anticipated temperatures are not unusually high for some conventional petroleum oils, it may be desirable to select a lubricant capable of operating at higher temperature in order to achieve a longer service life or some specific property. It may be possible to provide a lubricant that only requires changing every two years or even the life of the engine as is the case for oils in large steam turbines. A trade-off can be made between the cost of the lubricant and the service life.

Hydrocarbon oils that are extracted from petroleum sources and synthetic lubricants will be considered for use as the crankcase lubricant in the steam expanders. In the selection of a suitable lubricant, it is necessary to: first, select a suitable base and second to select the proper chemical additives that are compatible with the base and the environment. To be considered a good lubricant for the intended application, the lubricant must have the following important characteristics:

1. Good oxidation resistance
2. High thermal stability
3. Low volatility
4. Suitable viscosity
5. Low pour point
6. Adequate lubricity
7. High ignition temperature
8. Good hydrolytic stability

The last characteristic is particularly important with respect to steam expander applications in that some contamination of the crankcase with water is highly probable, particularly during start-up and shutdown. For this reason, the lubricant needs better rust protection and much better demulsibility than do current motor oils. Additives similar to those used in hydraulic oils, i.e., calcium sulfonates could be used effectively for rust protection. The partial organic acid esters and phosphorus containing acid esters also are used for this purpose. Rust inhibitors usually have a high polar attraction toward metal surfaces and form a continuous protective film over the surfaces. Care must be exercised in the selection of rust inhibitors to avoid corrosion reactions with nonferrous metals or emulsions with water. Chemical additives are rarely effective in improving the hydrolytic stability of lubricants and are seldom used to improve this property; hydrolytic stability is an inherent quality of the base lubricant. Hydrocarbons have excellent hydrolytic stability and major concern is with the additives and with synthetic lubricants. The straight alkyl-chain compounds generally are more easily attacked by water than the highly branched structures. For example, the esters have long chains and generally have poor hydrolytic stability at temperatures of 400°F and for this reason the esters will not be considered as a base for this application. Hydrolysis and water contamination can result in the following reactions: (1) change in physical properties, (2) generation of sludges and other insoluble compounds, (3) decrease in solubility of essential additives, (4) corrosive attack, (5) release of volatile compounds due to chemical breakdown. Obviously all of these changes are detrimental and must be kept to a minimum. It is expected that a complete reformulation of the crankcase lubricant will be required because most of the extreme pressure agents and inhibitors currently in use have little tolerance to water.

Because the bulk temperature of the crankcase lubricant is expected to be higher than the temperatures in the current internal combustion engines, oxidation stability of the lubricant must be superior to the current premium-grade motor oils. Oxidation of the hydrocarbons results in the formation of oil-soluble acid compounds which increases the vixcosity of the oil and can be corrosive; oxidation of the synthetics are more complex depending upon the type of structure, i.e., some form volatile gases and do not change the viscosity. Additives to inhibit oxidation include sulfur and phosphorus compounds and the amine and phenolic compounds.

Other additives which are believed to be required in the lubricant are defoamants and anti-wear additives. The most common defoamant is the organo silicon oxide polymer and only small concentrations are required (1 - 20 ppm) to inhibit foam formation caused by the action of the engine crankshaft. Anti-wear additives are required to minimize friction and wear in highly loaded components such as the cam/tappet interface where Hertzian stresses can be as high as 200,000 psi. Extreme-pressure (EP) additives are used and they react with the rubbing metal surfaces to form a lubricating film which protects the metal surface when the lubricating oil film is lost. The most common EP additives are zinc, phosphorus, sulfur and chlorine compounds.

In the survey for a suitable crankcase lubricant for the steam expanders to be tested in this program, one of the more attractive solutions appears to be the use of ahighly refined turbine oil. Many of these oils are the result of extensive development work at the Pennsylvania State University, where processes have been developed for "super-refining" mineral oils to achieve lubricants which have lower pour points, better viscosity-temperature characteristics and higher thermal and oxidative stability. Within the family of super-refined mineral oils, it is possible to use either paraffinic or aromatic-base oils. There are distinct differences in the physical properties of these two types of hydrocarbons. Because of intermolecular spacing, the paraffinic oils are more compressible, show less change in viscosity with temperature, have lower densities and lower pour points.

The production of super-refined mineral oils is essentially a clean-up operation where the oils are carefully dewaxed, many of the polar impurities are removed, and selective catalytic hydrogenation is used to minimize unsaturation. This process removes not only undesirable impurities, but also naturally-occurring oxidation inhibitors and polar compounds, which are essential for boundary lubrication. Thus, it is necessary to compound these oils with oxidation inhibitors and anti-wear additives. Since thermal decomposition will also proceed by the formation of free radicals, a small percentage of an aromatic disulphide was found to be effective in protecting the oil. The use of the aromatic disulphide gives an added bonus since it will also act as an anti-wear agent by surface reaction between the metal and the sulphide.

The use of blending techniques opens up many possibilities as far as the control of properties such as viscosity and pour point. A wide range of properties can be obtained by blending suitable base stocks to achieve the bulk viscosity and pour point required for the application.

A common turbine oil that has proven to have satisfactory service (∿ 20 years) at temperatures < 250°F in steam turbines and generators is produced by a number of refining companies and is known as Teresso 65, Industrial Oil 61, Tellus 69 or DTE Extra Heavy. The oil has oxidation and rust inhibitors but no EP anti-wear additive which would have to be formulated. Characteristics of the oil are as follows:

Flash point, deg F, min	420
Viscosity, Saybolt Universal, secs 100 deg F, min	540
max	700
Viscosity index, min	85
Pour point, deg F, max	+25
Neutralization value (total acid number) mg KOH/g, max	0.80
Oxidation stability test, hours, min (a)	1000
Sludge, %, max (b)	0.10
Rust prevention test	Shall Pass

(a) Test carried out to total acid number of 2.0 mg KOH/g.

(b) In oil taken from oxidation test.

The major disadvantages to the use of this oil are the relatively high pour point and the low hot spot temperature capability. Although a modified turbine oil would probably be satisfactory for the crosshead piston design, it may not be satisfactory for the trunk piston design because of the anticipated 540°F hot spot temperature.

The synthetic hydrocarbon lubricants have about a 100°F advantage over the mineral oil based turbine oils and appear to have the necessary properties for the application. For relatively long time service, the following generalized maximum service temperatures can be stated:

Base	Max. Bulk Oil Temp., °F	Max. Hot Spot Temp., °F
Mineral Oil	250° - 275° (possibly 300°F)	∿ 500
Synthetic Hydrocarbon	350° - 375° (possibly 400°F	∿ 600

A special synthetic hydrocarbon formulation (XRN-1301-C) was prepared by Mobile Research and Development Corp. for evaluation in the program. The lubricant has suitable oxidation, rust and foam inhibitors and a anti-wear additive. The properties are tiven in Table 4-7. The lubricant has a low pour point (< -65°F) and a viscosity in the SAE 30 range. However, its high viscosity index (147) puts it in the SAE 40 range at elevated temperature. The lubricant appears to have good antifoam performance and good rust protection and although it forms a small amount of emulsion, no emulsion problem would be expected in the engine.

A general comparison of various other synthetic lubricants is made in Table 4-8(44). Known disadvantages of some of these synthetic lubricants include poor hydrolytic stability of the esters and poor viscosities and high pour points of the polyphenyl ethers (in addition to their high cost). Projected costs of the synthetic hydrocarbons are not expected to exceed current premium grade motor oils.

The ESSO Research and Engineering Co. (Govt. Research Lab) is under contract to Steam Engine Systems Corp. to develop a suitable lubricant for the lubrication of cylinder walls in a reciprocating steam expander operating at 1000°F(45). In their work they found the inhibited blends of hydrorefined paraffinic distillates and residua to be the most suitable

Table 4-7

Properties of Experimental Synthetic Hydrocarbon Lubricant[a]

XRN 1301 C

Gravity, °API	37.4	
Specific Gravity	0.8378	
Pour Point, °F	< -65	
Flash Point COC, °F	460	
Viscosity, cs @ 100°F	75.45	
SUS @ 100°F	349.7	
cs @ 210°F	11.07	
SUS @ 210°F	63.1	
Viscosity Index	147	
Acid No. ASTM D-664-1	0.06	
Base No. ASTM D-664-3	0	
ASTM Foam Seq. I Tendency, ml	350	
Stability, ml	0	
Seq. II Tendency, ml	40	
Stability, ml	0	
Seq. III Tendency, ml	410	
Stability, ml	0	
ASTM Rust, Distilled Water	Pass, Pass	
ASTM Rust, Sediment Water	Pass, Pass	
Emulsion ASTM D-1401		
Total Emulsion, ml	5	7
Time for 3 ml, min.	32	31
Time for complete break, min.	35	37

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[a] Mobil Research and Development Corp.

Table 4-8

Performance of Selected Types of Synthetic Oils[(a)]

Class of Compound	Probable Maximum Thermal Stability Limit, °F	Resistance to Water Degradation	Extreme Pressure Lubricity
Polyglycols	600	Excellent	Good
Phosphate esters	800	Good	Excellent
Dibasic acid esters	600	Good	Good
Chlorofluorohydrocarbons	600	Excellent	Excellent
Silicones	900	Poor	Poor (Steel)
Silicate esters	800	Poor	Fair
Fluoroesters	600	Good	Poor
Neopentyl polyol esters	600	Good	Good
Polyphenyl ethers	600	--	Good
Silanes	700	--	Poor

(a) Reference 44.

of the petroleum base oils. A combination tricresyl phosphate (TCP) and ethyl-702 appears to offer the best anti-wear and anti-oxidant properties in the selected bases. The candidate lubricant (designated 2415-50-1) has a higher viscosity (SUS) rating at the standard temperatures of 100°F and 210°F, 663 and 74 respectively, but a lower viscosity index (102) that the Mobil XRN-1301-C lubricant. However, as discussed previously, there is no requirement to lubricate the cylinder walls or valve parts in the General Electric program.

4.3 Lubricant Selection Study

In the steam expander design, there are three areas requiring study of lubrication requirements including the inlet valve, the piston/cylinder and the shaft connecting the piston with the crosshead piston. The inlet valve stem will slide in a guide that is exposed to steam and that is sealed by means of compression rings. The most severe lubrication problem is the piston/cylinder wall; in this case the power piston rings will serve as the lubricant. The shaft connecting the steam piston to the crosshead piston will slide in a seal that is exposed to steam on one side and oil on the other. Each of these areas require proper design and material selection.

Although the qualitative correlation of the coefficient of friction and wear rate is generally recognized, the exact correlating function between the two is difficult to obtain. Theoretical relationships may be developed which are useful in ranking material and lubricant combinations. However, the reduction of the theoretical relationships to useful design parameters by testing at the expected service conditions is necessary.

Friction effects arise from the tangential forces transmitted across the interface of contact between two bodies. The wear phenomena consist of the removal of material from the contacting surfaces. Adhesion, the ability of contacting bodies to withstand tensile forces after being pressed together, is the primary interacting phenomena. The concept of surface energy makes it possible to develop parameters for making rudimentary predictions of the performance of specific materials under sliding conditions.

While 90 percent or more of the resistance to sliding arises from the need to shear strongly adherent surface atoms of the contacting materials, other factors have to be considered. A roughness component arises from the necessity of lifting one surface over the other's asperities, which usually contributes about 0.05 to the friction coefficient. Hydrodynamic lubrication can eliminate this component (liquid phase lubrication). A plowing component is present when a hard, sharp surface digs into a softer surface and produces a groove. For really rough surfaces this term may be large, but otherwise it is usually negligible.

Adhesive wear is best combatted with hard materials with low interaction (or one member of the pair a non-metal), or by use of two metals with high interaction reduced by a good boundary lubricant. The use of hard materials does not in itself produce significantly lower wear rates, but the probable accompanying change in chemical structure may produce factor-of-10 changes in the wear rate.

By using a model for solid lubrication which assumes solid-solid contact over the fraction x of a junction, with solid film lubrication over the rest, the resultant coefficient of friction is (m = metal, 1 = lubricant):

$$f = \alpha f_m + (1 - \alpha) f_1$$

If it is assumed that no wear of the materials occurs in the lubricated region, the area of breakthrough is circular, the particle produced is helispherical, and the probability of producing a wear particle is the same for the penetrated part of the junction as it is for the unlubricated materials:

$$\frac{\text{Wear lubricated}}{\text{Wear unlubricated}} = \beta = \frac{d^3}{d_m^{\ 3}}$$

And since $$\alpha = \frac{d^2}{d_m^{\ 2}}$$

$$\therefore \alpha = \beta^{2/3} \quad \frac{f - f_1}{f_m - f_1}$$

which correlates friction and the wear ratio. The equation suggests that the effect of a lubricant on the wear will be much greater than its effect on the friction.

Although adhesive wear (sometimes called galling and scuffing) resulting in welding of the surface asperities is usually associated with metallic materials, the adhesion theory is also applied to solid polymers. The difference between the metals and plastics is that the deformation in plastics may be partly elastic over a wide range of load. Also, the flowing term and surface roughness are of greater importance in plastics. The mechanism of wear with carbon-graphites is primarily abrasion which occurs when asperities of two moving surfaces touch and wear fragments are formed from one or both surfaces. Grooves are generally plowed in the "softer" carbon graphite.

Candidate solid lubricants are to be avaluated in a reciprocating wear test rig prior to their selection for further evaluation in the single cylinder engine tests. The wear test rig will simulate all conditions in the engine with the exception of steam pressure, i.e., 1000°F steam temperature, up to 500 psi side load and an average linear velocity of 1200 ft/min. The wear test rig design has been completed and the facility installed and ready for check out at the GE Large Steam Turbine and Generator Dept. at Schenectady. The specimen design consists of a reciprocating 1.990 ± .001 inch OD x 4-3/4 inch long cylinder rubbing against a stationary 0.20 inch thick x 0.70 inch wide specimen that is contoured to mate the reciprocating 1.990 inch diameter surface. A schematic of the test rig is shown in Figure 4-2 and the initial installation is shown in Figure 4-3.

The candidate materials that were selected for evaluation in the wear test rig are listed in Table 4-9. The material combinations to be tested are listed in the test matrix in Table 4-10.

4.4 Lubricant Recommendations

4.4.1 Solid Lubricants

Based on the current technology of solid lubrication and wear resistant materials that was discussed in Sections 4.1 - 4.3, a list of

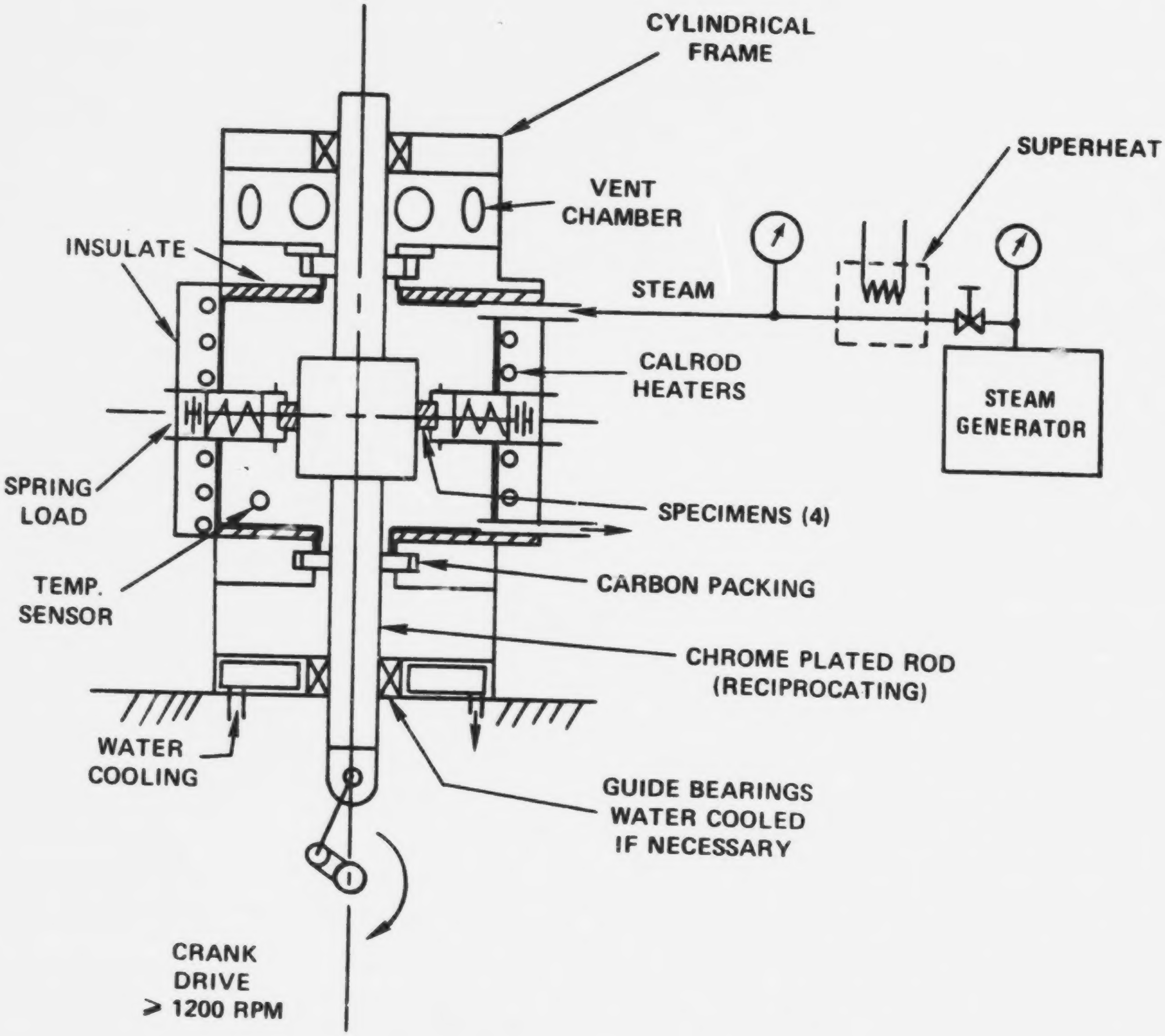

CYLINDRICAL FRAME
SUPERHEAT
VENT CHAMBER
INSULATE
STEAM
CALROD HEATERS
STEAM GENERATOR
SPRING LOAD
SPECIMENS (4)
TEMP. SENSOR
CARBON PACKING
CHROME PLATED ROD (RECIPROCATING)
WATER COOLING
GUIDE BEARINGS WATER COOLED IF NECESSARY
CRANK DRIVE
> 1200 RPM

Figure 4-2. Solid Lubricant Wear Test Rig.

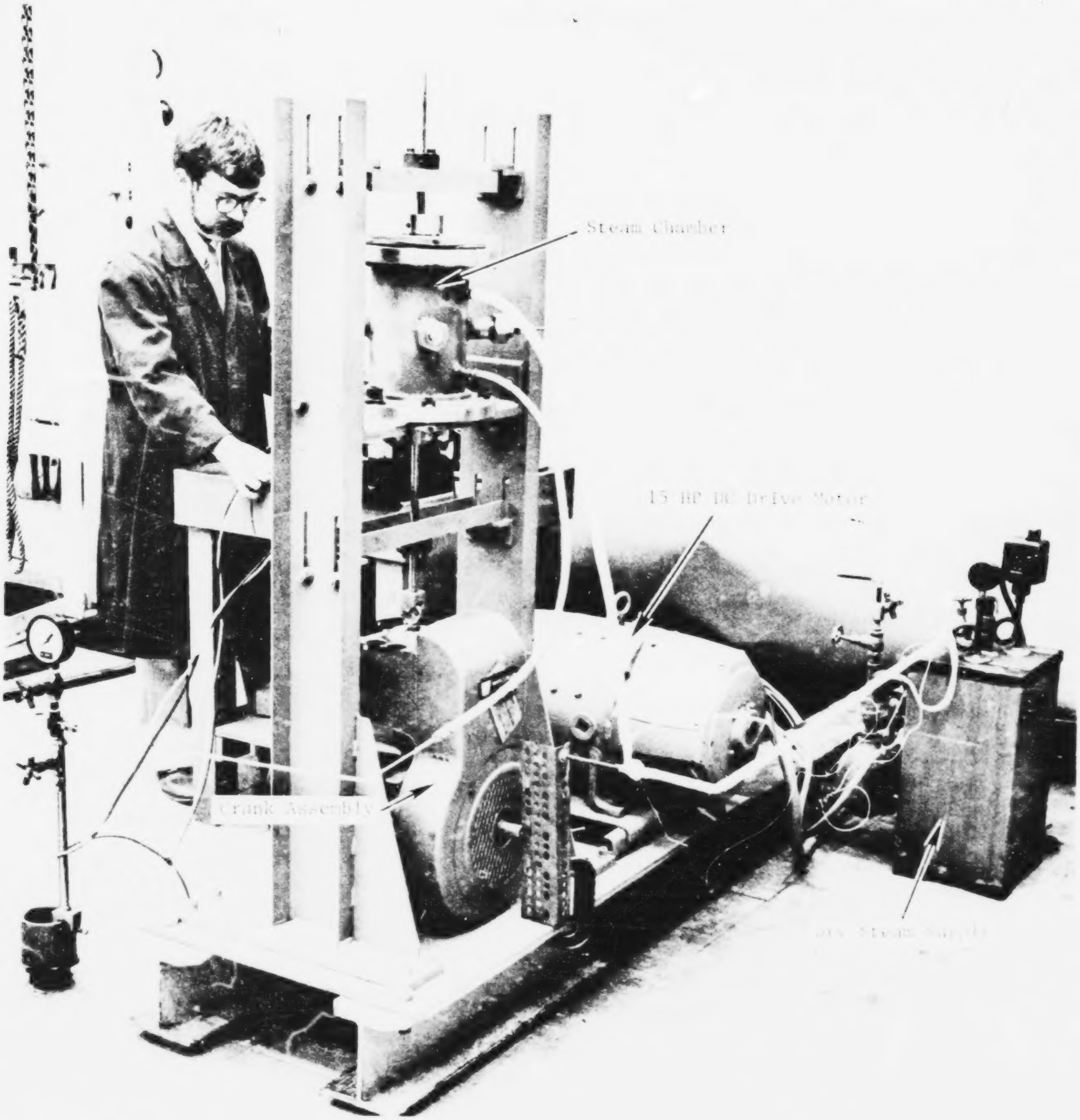

Steam Chamber
15 HP DC Drive Motor
Crank Assembly
Dry Steam Supply

Figure 4-3. Wear Test Rig Installation.

Table 4-9

Wear Test Materials

Ring Specimens

	Material Class	Manufacturer	Grade
1.	Carbon-Graphite	US Graphite	107 (Sb Impregnated)
2.	Carbon-Graphite	US Graphite	2980
3.	Carbon-Graphite	Pure Carbon	P-03
4.	Carbon Graphite	Pure Carbon	P-5NR
5.	Carbon Graphite	Carbone	5890
6.	Carbon-Graphite	Carbone	5890 & Ceramic
7.	Carbon-Graphite	Carbone	JP-500
8.	PTFE	Modern Ind Plastics	15% Bronze + 5% MoS_2
9.	PTFE	Modern Ind Plastics	20% Glass
10.	PTFE	Modern Ind Plastics	10% Carbon
11.	Porous Metal	Astromet	Inconel 600 + (BaF_2 + CaF_2)
12.	Porous Metal	Astromet	Inconel 600 + PTFE
13.	Porous Metal	Astromet	Inconel 600 + Graphite
14.	Metals	Metsubishi	Bishiralloy C (Pb Impregnated Fe Alloy)
15.	Metals	Koppers	K-35 (Ni-Resist)
16.	Hard Surfacing	Koppers	K-1051 over Inconel X
17.	Hard Surfacing	DuPont	LPA-100 + 20% Ni (Laves Phase 65 Vol. %)over AISI 410 SS
18.	Hard Surfacing	DuPont	Ni_3B over AISI 410 SS
19.	Hard Surfacing	UCC	LSR-1 (TiC) over AISI 410 SS
20.	Hard Surfacing	UCC	LW-1 (Co Bonded WC) over AISI 410 SS
21.	Hard Surfacing	UCC	LC-1 (Cr_3C_2 + 15% NiCr) over AISI 410 SS

Table 4-9 (Cont'd.)

Wear Test Materials

Cylinders

Material Class	Manufacturer	Grade
1. Carbon-Graphite	Carbone	5890
2. Metal	-	AISI 440C
3. Metal	-	Cr Plate over AISI 410 SS
4. Hard Surface	UCC	LC-19 (Cr_2O_3 + Al_2O_3) over AISI 410 SS
5. Hard Surface	UCC	LC-1 (Cr_3C_2 + 15% NiCr) over AISI 410 SS
6. Hard Surface	UCC	LSR-1 (TiC) over AISI 440C SS
7. Hard Surface	Armalloy	Chromize over AISI 440C SS

Table 4-10

Wear Test Program Matrix

Test No.	Ring Material	Cylinder Material	Temp., °F
1	1 2	2	750
2	3 4	2	750
3	5 6	2	750
4	7 12	2	750
5	8 9	3	500
6	10 12 or 14	3	500
7	11 13	2	750
8	17 18	2	750
9	20 21 or 16	4	750
10	17 18 or 15	4	750
11	5 21	1	750
12	1 3	5	750

NOTES:
1. Avg. linear velocity, 1200 ft/min.
2. Side load 300 psi
3. Test duration, 150 hours
4. Replication, 2

candidate Piston Ring/Cylinder Liner combinations was compiled and is tabulated in Table 4-11. The primary selection at this time is a metal impregnated carbon-graphite material for the piston rings rubbing against AISI 440C SS or hard chromium plated steel cylinder liner. The same material combination was selected for the back-up design for the piston rod seal in the crosshead piston design. The primary selection is a bronze filled PTFE rod seal material. A hard surfacing material (LPA-100 + 20% Ni) and Ni-Resist (D-3 type) cast iron materials combination was selected for the valve stem ring/valve guide components.

4.4.2 Liquid Lubricants

Although a final selection has not been made for the crankcase lubricant, the Mobile Synthetic hydrocarbon XRN-1301-C has been tentatively selected. This oil has a low pour point, good thermal stability and oxidation and rust inhibition and good viscosity ratings.

Table 4-11

PISTON RING/CYLINDER LINER COMBINATIONS

	RING	CYLINDER LINER
1.	K-35 (Ni-Resist)	Hard Cr Plate(b)
2.	K-35 (Ni-Resist)	Hard Cr Plate(b) + MoS_2 Bonded Film
3.	K-35 (Ni-Resist)	Hard Cr Plate(b) + $(CF_x)_n$ Burnished Film
4.	Carbon-Graphite, Grade 107(a)	Hard Cr Plate(b)/Type 440C SS
5.	Hast X Porous Structure + Graphite Filled	Hard Cr Plate(b)
6.	Koppers 1051 over Inconel X 750	Type 440C SS
7.	LPA 100 + 20 Ni over Inconel X 750	Hard Cr Plate(b) Nitride Ni-Resist D2/D3
8.	Hast X Porous Structure + NBS 418 + (CAF_2 + BaF_2)	Hard Cr Plate(b)
9.	LW-1 (WC)	Cast Iron LC-19

(a) Primary Recommendation for first engine.

(b) Limited to temperature below 700°F; for temperatures in excess of 700°F, one of the following liner coatings can be used:

- LC-19 (CR_2O_3 + Al_2O_3)
- LC-1 (Cr_3O_3 + 15 NiCr)
- Chromize (i.e., Armalloy)

5.0 TASK IV
FABRICATION AND TEST

5.1 Crosshead Expander Manufacturing Drawings

All manufacturing drawings for the crosshead single cylinder steam expander were completed and issued for fabrication by December 23, 1971. Also, all drawings required for the modification of a CFR-48 crankcase, manufactured by The Waukesha Motor Co., were completed and issued prior to December 19th. The main modification to be made to the CFR crankcase is the addition of an idler gear between the crankshaft and camshaft gears to provide a 1 to 1 speed ratio between the crankshaft and camshaft. This modification will make use of existing gears and gear stub shaft and requires only minor machining of the crankcase. Other required modifications to the crankcase involve the addition of oil lines and oil jets.

Figure 5-1 (Dwg. 707E713 Sh. 1) shows an overall view of the crosshead expander. As indicated above the steam expander is designed to make maximum use of an existing crankcase. Therefore the details of the crankcase are not shown by Figure 5-1.

The crosshead expander consists of an oil lubricated aluminum crosshead which is connected to the steel power piston by a rigid connecting rod. Isolation of steam and oil is accomplished by a rod seal between the crosshead and the power piston. Two different seal types are shown and both seal types are being procured. The segmented carbon seal will serve as a backup for the impregnated Teflon chevron seal. Both rod seals include an oil exclusion seal. The power piston connecting rod is tubular for reduced weight, and to permit internal oil cooling. An oil jet located in the top of the crosshead wrist pin will spray oil into

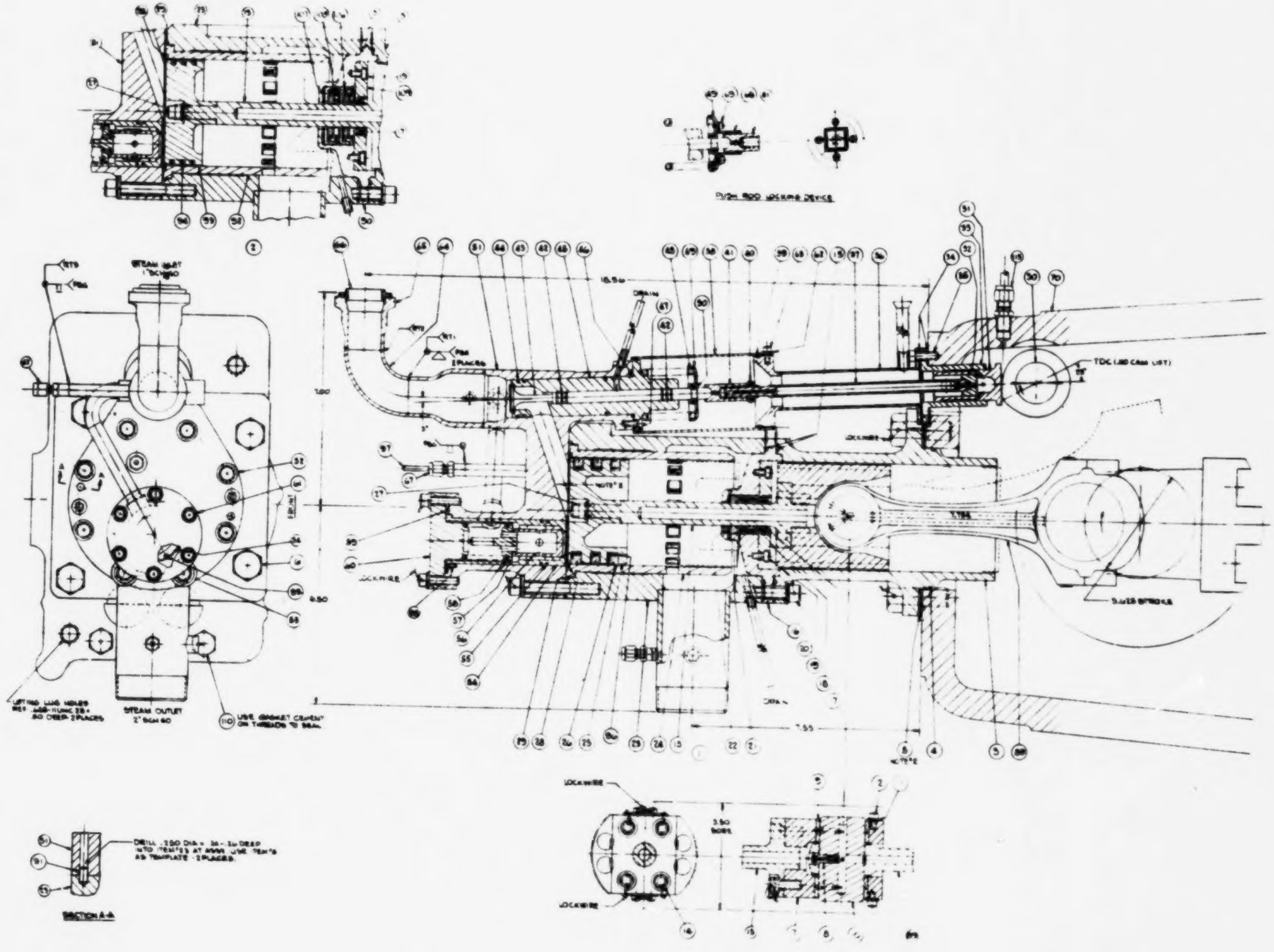

STEAM OUTLET
SECTION A-A
PUSH ROD LOCKING DEVICE
LOCKWIRE
LOCKWIRE
DRAIN

Figure 5-1. Steam Expander Assembly - Crosshead Type Piston (GE Dwg. 707E713, Sh. 1).

the power piston connecting rod primarily to remove heat generated by the connecting rod seal.

The crosshead wrist pin and wrist pin bearing design received much attention as discussed under Task I above. The wrist pin bearing design shown by Figure 5-1 is quite similar to current 2 cycle diesel engine practice. Oil is supplied to the bearing under pressure, and the bearing contains closely spaced helical grooves to permit the free flow of oil to all surfaces of wrist pin.

Two power piston designs are shown so that piston rings of different materials can be evaluated. The basic expander design shows three relatively wide rings which are pressure balanced segmented graphite rings with an Inconel inner (spring) ring. The alternate power piston and ring design provides for the evaluation of metal rings such as chrome carbide coated Inconel-X. The expander has a removable cylinder liner so that appropriate piston ring and cylinder materials can be combined with minimum difficulty.

A number of valve and valve gear configurations were studied. The L-head poppet valve configuration shown by Figure 5-1 was concluded to best meet all design requirements. Of great importance, lubricating oil is well isolated from the high temperature parts of the expander - thus eliminating the need for thermal insulation to prevent high oil temperature. Also the orientation of the valve gear minimizes the risk of oil entering the steam system by way of the steam inlet valve.

The inlet valve assembly is designed so that the repair or interchange of critical components such as valve stem seals and valve seat can be accomplished with minimum difficulty. For example different material combinations for the valve and valve seat, and for the valve stem bearing and seals can be evaluated merely by insert replacement without remachining of parts.

In the design of the valve gearing minimum mass and linkage flexibility was emphasized to provide fast valve response and to minimize inertia forces throughout the valve gear system. As discussed under Task I above, valve gear stresses, including cam Hertzian stress, are well within accepted practices. The valve gearing is well oil lubricated except for the top portion of the valve stem which is provided with solid lubrication. Although the valve cam will receive oil by splash lubrication, an oil jet is provided to insure sufficient lubrication between the cam and cam follower.

A recompression relief valve is provided in the cylinder head. This valve is designed to relieve the cylinder pressure under all conditions of expander operation, including motoring, before the steam inlet valve is forced open due to excessive cylinder pressure.

5.2 Trunk Expander Manufacturing Drawings

All manufacturing drawings for individual components for the trunk piston single cylinder steam expander have been completed and issued for fabrication. A few assembly drawings remain to be completed. The trunk piston steam expander also uses the CFR-48 Waukesha crankcase. Modification drawings for the crankcase have been completed. Modifications to the crankcase for the trunk piston are essentially the same as for the crosshead expander except for minor variations in dimensions.

Figure 5-2 (Dwg. 707E719 Sh. 1) shows an overall view of the trunk piston expander. Except for the power piston the trunk piston expander is quite similar to the crosshead piston expander. For example there is no difference in basic design for the valve gearing or recompression valve for the two expanders. Even the connecting rod wrist pin and wrist pin bearing designs are essentially the same for the two expanders. Two major differences between the two expanders are: the trunk piston expander is potentially shorter in overall height, and the power piston and connecting rod assemblies are quite dissimilar. However, the reciprocating weights for the two expanders are almost the same - approximately 8.2 pounds.

The trunk piston is a two piece design which permits the following:

1. The use of the same wrist pin design as used in the crosshead expander.
2. The use of dissimilar materials for the top and bottom portions of the piston for weight reduction.

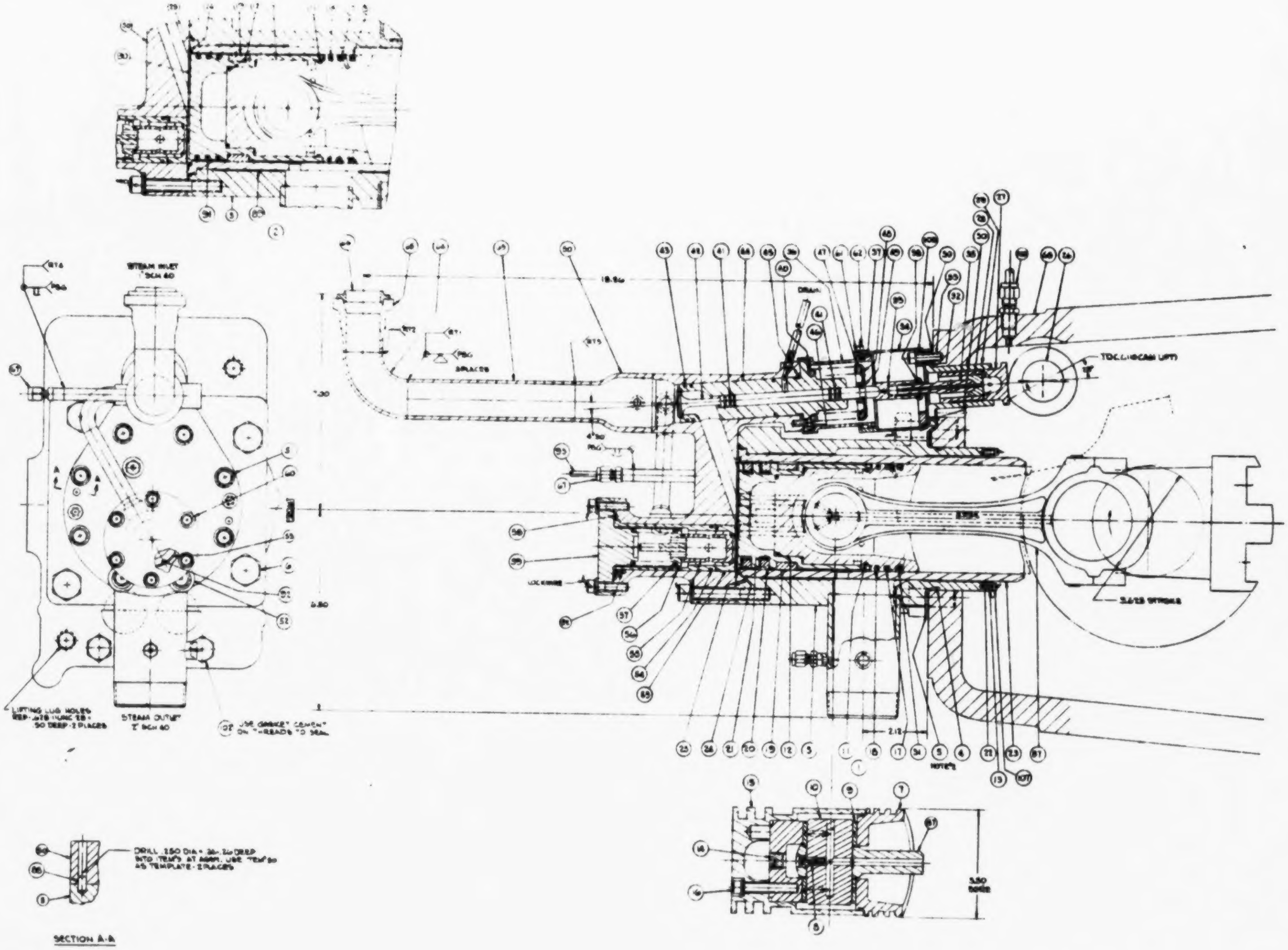

STEAM INLET
1" SCH 80
STEAM OUTLET
2" SCH 80
LIFTING LUG HOLES
USE GASKET CEMENT ON THREADS TO SEAL
DRILL .250 DIA
AS TEMPLATE - 2 PLACES
SECTION A-A
DRAIN
TDC
STROKE
19.50
7.30
6.80
2.12
3.50
57
5
60
53
6
52
52
64
86
8

Figure 5-2. Steam Expander Assembly - Trunk Type Piston (GE Dwg. 707E719, Sh. 1).

3. Provides for positive oil/steam isolation in the region of the wrist pin while permitting adequate oil lubrication of the wrist pin without subjecting the oil to high temperature.
4. Provides for oil cooling of the lower portion of the piston.
5. Provides for the interchange of the upper portion of the piston to accommodate alternate sealing ring configurations and materials.

To keep the trunk piston as short as possible, the basic expander design shown in Figure 5-2 shows two non-metallic sealing rings compared with the crosshead expander, Figure 5-1, which contains three non-metallic sealing rings. However, the alternate trunk piston design contains three metallic sealing rings for the same piston length. A number of sources indicate that from one to four sealing rings are required. Tests on IC engines indicate that two rings are sufficient if both function properly. A third ring (a rider ring) is required in the top portion of the trunk piston to carry piston side loads and is not intended to function as a seal ring.

The trunk piston contains three rings near the bottom of the piston. The top ring of the three is a seal ring to restrict the flow of steam into the expander crankcase. The two bottom rings are oil exclusion rings to present oil from entering the steam system. The lower surface (piston skirt) of the bottom portion of the trunk piston functions as a piston support or second rider ring. Since the lower part of the trunk piston is oil lubricated, a separate solid lubricant rider ring is not required below the wrist pin.

The trunk piston expander also has a removable cylinder liner so that different ring/liner material combinations can be evaluated with minimum expander rework.

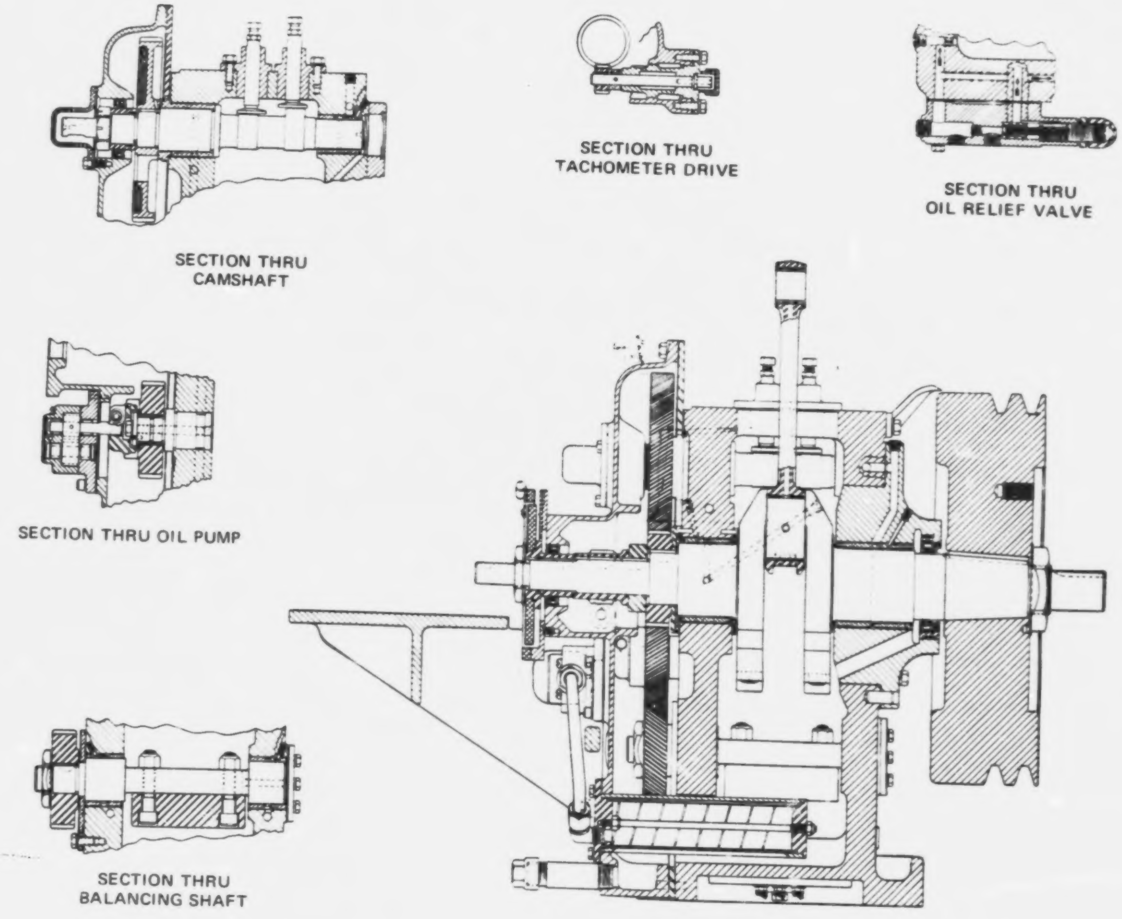
5.3 Crankcase Design

In order to minimize steam expander cost and fabrication time existing single cylinder, reciprocating, internal combustion engines were studied for the purpose of utilizing their basic crankcase assembly as a starting point for the steam expander design. Major factors considered in the study were crankcase ruggedness including crankshaft and bearing high load capacity. The study was narrowed down to two engine manufacturers, these being The Waukesha Motor Company and the Laboratory Equipment Company (LEBACO) both of which make heavy duty test engines of single cylinder configuration. Both companies were visited and their engines reviewed under fabrication, assembly and test. The Ford Motor Company was visited where both the Waukesha and LEBACO test engines have been used extensively for internal combustion engine development work. As a result of the study, The Waukesha Motor Company, Model CFR-48 crankcase assembly with their 3-5/8" stroke crankshaft, was selected for both the crosshead and trunk piston steam expander designs.

Figure 5-3 is a cross section of the Model CFR-48 crankcase assembly. With the exception of the camshaft, camshaft gear, connecting rod distributor and timing indicator, Figure 5-3 illustrates the basic assembly purchased from The Waukesha Motor Company.

The crankshaft has a 3.625" stroke, 3.00" diameter main journals and 2.50" diameter crankpin. All journals are nitrided and ground to an 8 RMS finish. Main bearings are bronze sleeve type pressure lubricated through oil gallies in the crankcase. Oil is fed to the crankpin through holes in the shaft from the front main bearing. Thrust is taken by slotted face type bronze bearings on each side of the front main bearing. The rear end of the shaft is tapered and is provided with a key way and thread for flywheel installation. Both front and rear ends of shaft are also keyed for accessory and power take off respectively.

A gear case bolted to the front of the casing contains all of the gearing for the camshaft, balance shafts, oil pump and take off of external accessories. Counter rotating balance shafts are driven at engine speed thru an idler gear. All gearing is in the same plane and helical type gears, cast iron running against steel, are employed. Two camshaft positions, right and left, are available. The standard camshaft position is on the right side (flywheel end looking forward) and a two lobe cam operating at half engine speed (four cycle) is normally utilized for internal combustion engine testing. The top of the case is bored for two tappet assemblies on the right side. Figure 5-4 indicates the bearing bores for the two camshaft positions (upper right and left), crankshaft



SECTION THRU CAMSHAFT

SECTION THRU TACHOMETER DRIVE

SECTION THRU OIL RELIEF VALVE

SECTION THRU OIL PUMP

SECTION THRU BALANCING SHAFT

Figure 5-3. Waukesha CFR-48 Crankcase Assembly
(Source Reference: Waukesha Motor Company Bulletin 850M).

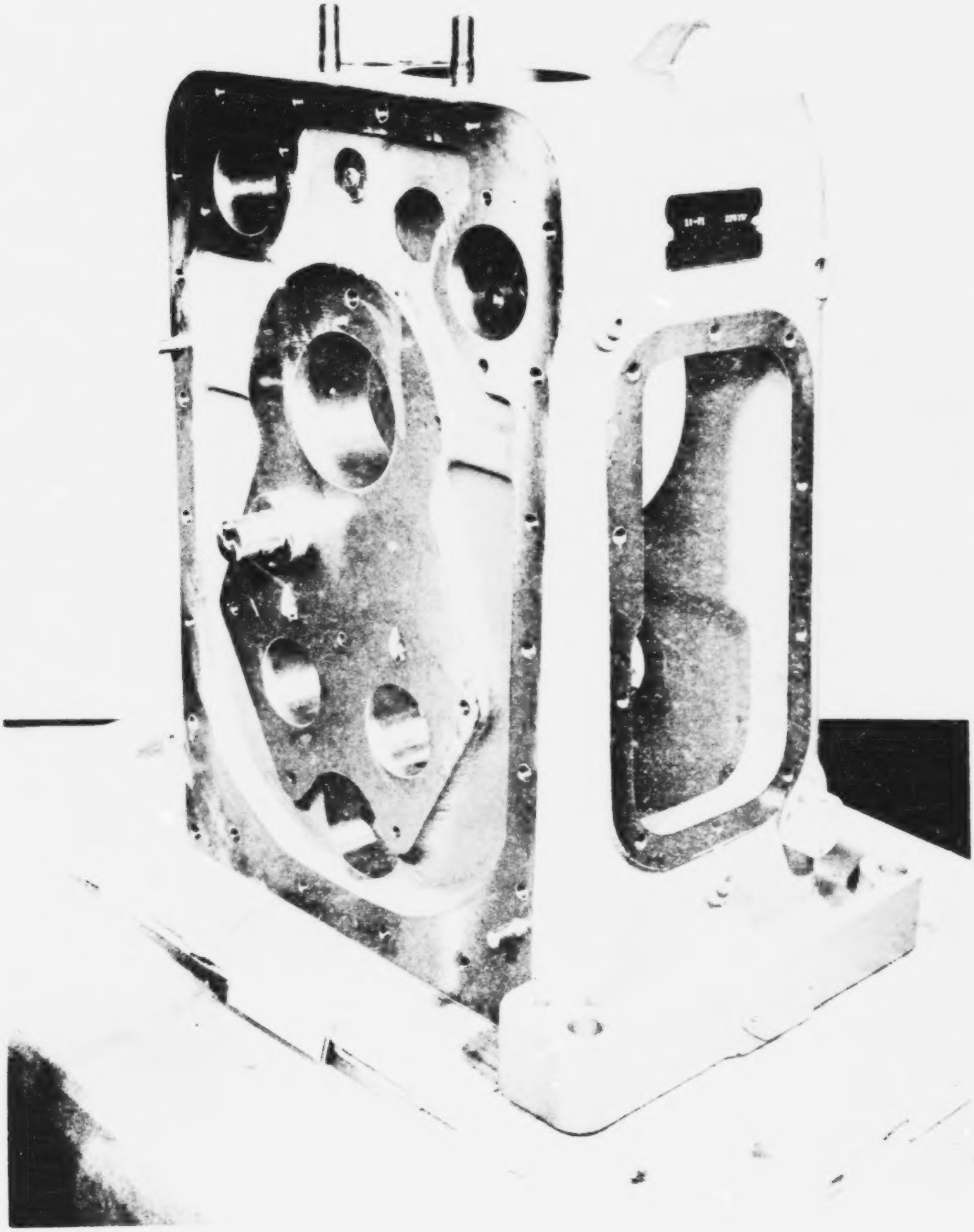


Figure 5-4. Waukesha CFR-48 Crankcase.

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(center), and dual balance shafts (lower left and right). An idler stub shaft protrudes below the crankshaft bore.

The lubrication system consists of a screen mounted in the case sump, a gear pump driven from the idler gear and a pressure regulating valve located at the rear right corner of the case assembly.

Oil gallies bored in the casing lead to all bearing bores providing full pressure lubrication to all bearings.

A 16" diameter steel flywheel with timing graduations at 1/2° intervals and two V-belt grooves completes the crankcase assembly as purchased from The Waukesha Motor Company.

Some modifications to the basic crankcase assembly are necessary and these are shown on Figure 5-5 (Drawing 221R901). Briefly these modifications are as follows:

1. Increase the size of the cylinder bore opening from 3-1/2" to 4-5/8". This was necessary to handle the 3-1/2" cylinder bore of the crosshead and trunk piston steam expanders.
2. Increase the size of cylinder to crankcase studs from 1/2" to 5/8". This is to provide for the higher than normal cylinder MEP as is normally done when the CFR-48 crankcase is adapted to diesel engine testing.
3. Plug the two right hand tappet assembly bores and provide one new tappet assembly bore on the left side. The left hand camshaft position is utilized on both expanders since greater clearance for the large single lobe camshaft was necessary. Provide an oil galley to the new tappet assembly bore.
4. Plug the forward and rear camshaft bearing bores on the right side to maintain integrity of the lube system.
5. Install a new idler shaft and idler gear between the crankshaft gear and camshaft gear to provide camshaft rotation at engine speed. Standard Waukesha CFR-48 parts will be used for this design. The idler stub shaft is identical to the lower stub shaft with the exception that oil porting was modified. The idler gear is identical to the idler which drives the dual balance shafts with its oil pump drive key adapter removed.

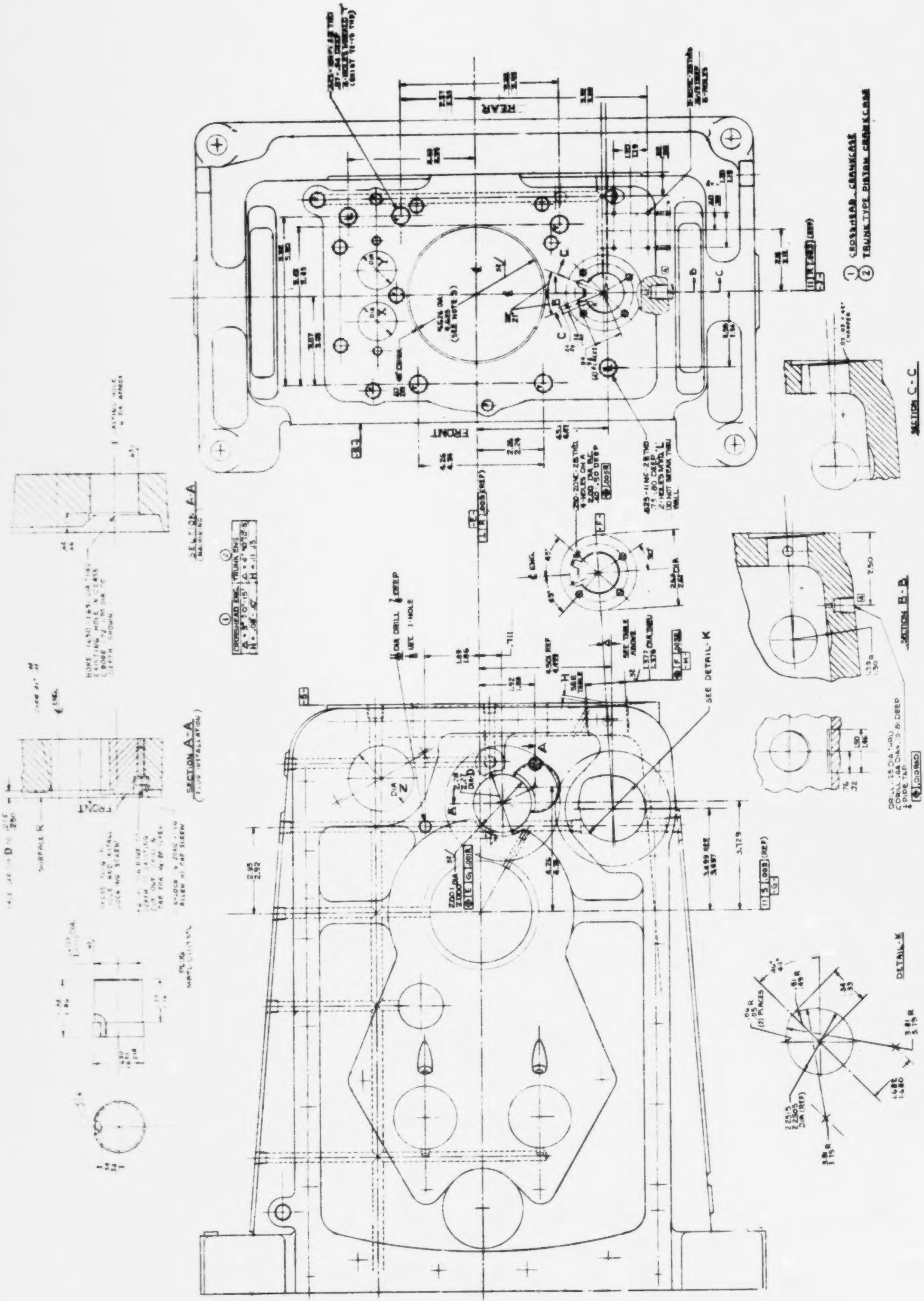
FRONT
REAR
SECTION A-A
SECTION A-A
SECTION B-B
SECTION C-C
DETAIL-K
SEE DETAIL-K
① CROSSHEAD CRANKCASE
② TRUNK-TYPE PISTON CRANKCASE
X
Y
Z
A
B
C
D
H

Figure 5-5. Steam Expander Crankcase, Modification Details (GE Dwg. 221R901).

The camshaft gear is identical to the crankshaft gear but contains a new bushing to fit the camshaft.

6. The left camshaft bore was opened up for 90° of its circumference so that the large cam lobe for the steam expander can be inserted. A relief inside the crankcase casting was also provided for lobe clearance. Camshaft bearings are standard CFR-48 bronze sleeve bearings.
7. Design and fabricate new balance shaft weights to match the cross head and steam piston design. This requires the use of a high density metal since space limits the physical size of the balance weights. Mallory Metallurgical Company alloy #1000 was selected for this. This material is 90% W, 6% Ni and 4% Cu with a density of 16.96 grams/cc. Final machining of the weights will not be completed until actual weight measurements of reciprocating hardware is obtained.
8. Small addition of weight to the crankshaft counterweight is necessary. This will be accomplished by adding shims under the counterweight attaching bolts. Connecting rod weight will be accurately measured before these shims are made.
9. Modified Waukesha bearings will be used for the crankpin. Bearings are #77 Trimetal fabricated by Clevite Corporation. A full 360° circumferential oil groove will be machined in these bearings to provide a continuous flow of oil to the steam expander wrist pin bushing.

Some modifications are being made to the crankcase lube oil system. A cartridge type oil filter, an oil cooler (water as coolant) and turbine type flow meter have been added. The present crankcase pressure regulating valve will control the amount of oil pumped to the main oil galley. Present plans are to hold oil pressure at 45 psig. Oil temperature will be regulated by the cooler water flow valve. Oil by-passed by the regulator will go directly to the crankcase sump without being cooled. Present plans are to hold sump temperature at approximately 250°F to vaporize water which might enter the oil sump. An electric calrod type heater mounted under the crankcase will be used to preheat the oil on startup. The crankcase breather pipe located on the right side of the engine will be piped to a small water cooled heat exchanger so that water vapor flowing from the crankcase can be condensed and measured.

Both Waukesha CFR crankcases have been received, and have been disassembled so that the above listed modifications can be made. Figure 5-6 shows the crankcase in the as received condition and Figure 5-4 shows the basic crankcase casting. Both crankcases are now in the shop where modifications are being made.

5.4 Test Planning

Instrumentation for the steam expanders will consist of the following: See Figure 5-7 (Drawing 707E700).

1. Horizontal and vertical vibration pickups located in the plane of the rear main bearing.
2. Magnetic pickup for determining TDC during operation. This will be read out on an oscilloscope.
3. Magnetic pick-up for speed. A six tooth gear will be attached to the shaft coupling.
4. Strain gage type dynamic pressure transducer located in the cylinder head. This unit will be water jacketed and rated for 1000 psi.
5. Strain gage dynamic pressure transducer located in the intake duct similar to Item 4.
6. Strain gage static pressure transducer located in the exhaust duct rated for 60 psia.
7. Inlet and exhaust temperature probes.
8. Approximately 30 thermocouples located on various parts of the expander to monitor temperatures.
9. Oil flow sensor, pressure gages and oil temperature at various points in the system.

5.5 Steam Expander Test Facility

A schematic representation of the steam expander test facility is shown in Figure 5-8 (Drawing 707E676). General arrangement of the Steam Expander Test Facility is shown in Figure 5-9 (Drawing 47R199790).

The steam generator is an electrically heated once-through boiler-superheater which is designed to provide 1000 psi, 1000°F, superheated

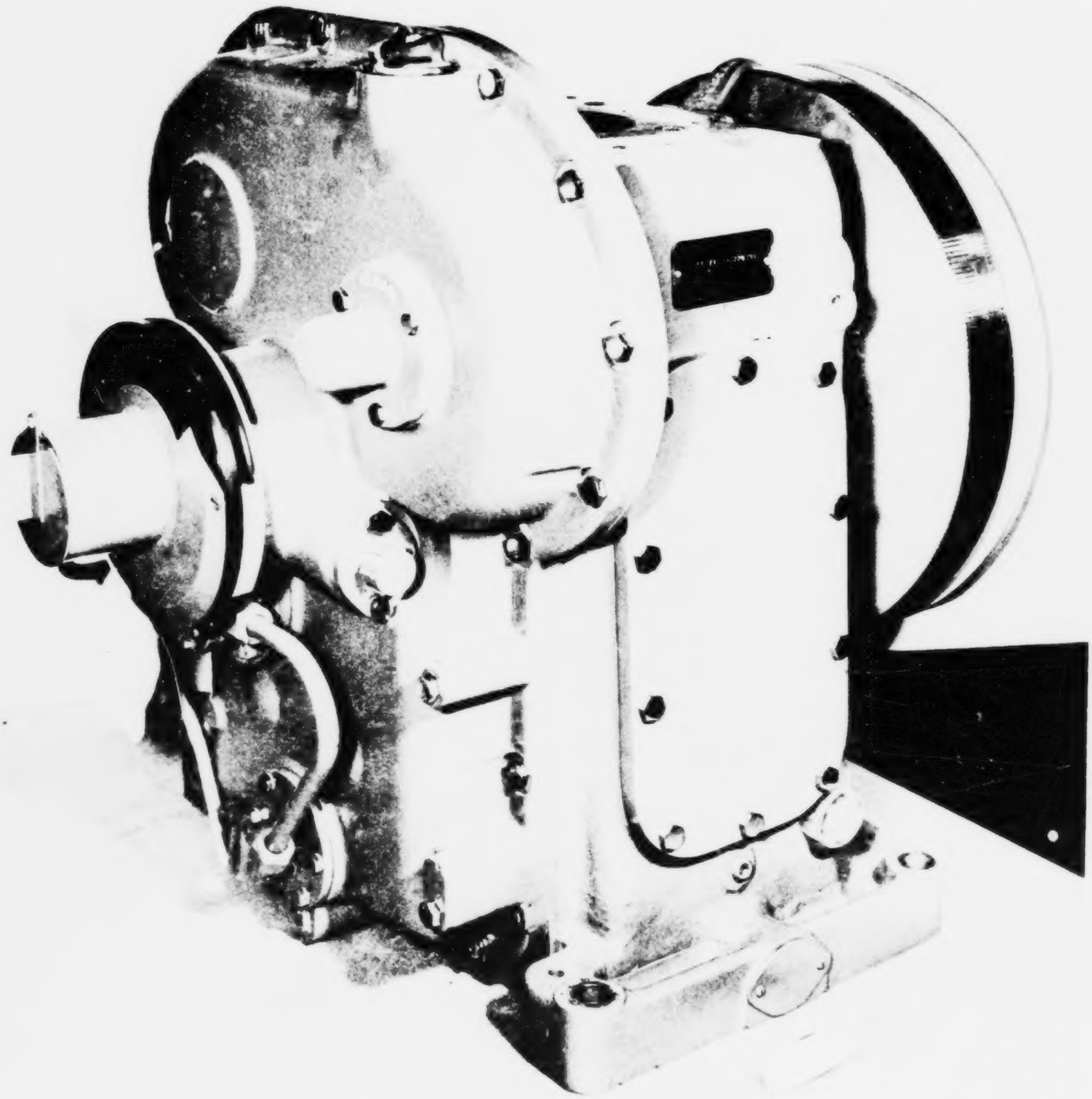

Figure 5-6. Waukesha CFR-48 Crankcase Assembly.

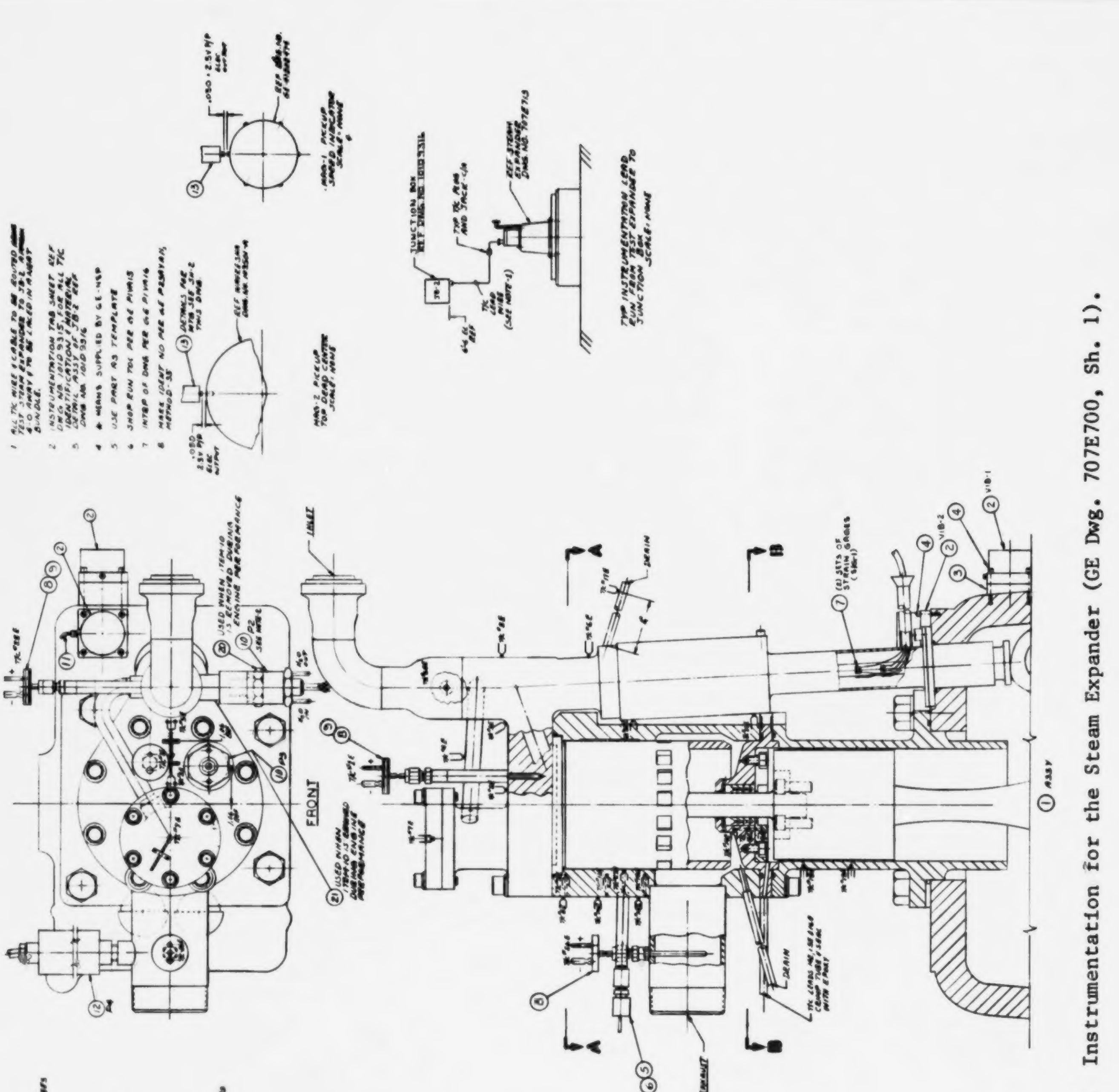
FRONT
INLET
EXHAUST
DRAIN
VIEW-AA
VIEW-BB
JUNCTION BOX

Figure 5-7. Instrumentation for the Steam Expander (GE Dwg. 707E700, Sh. 1).

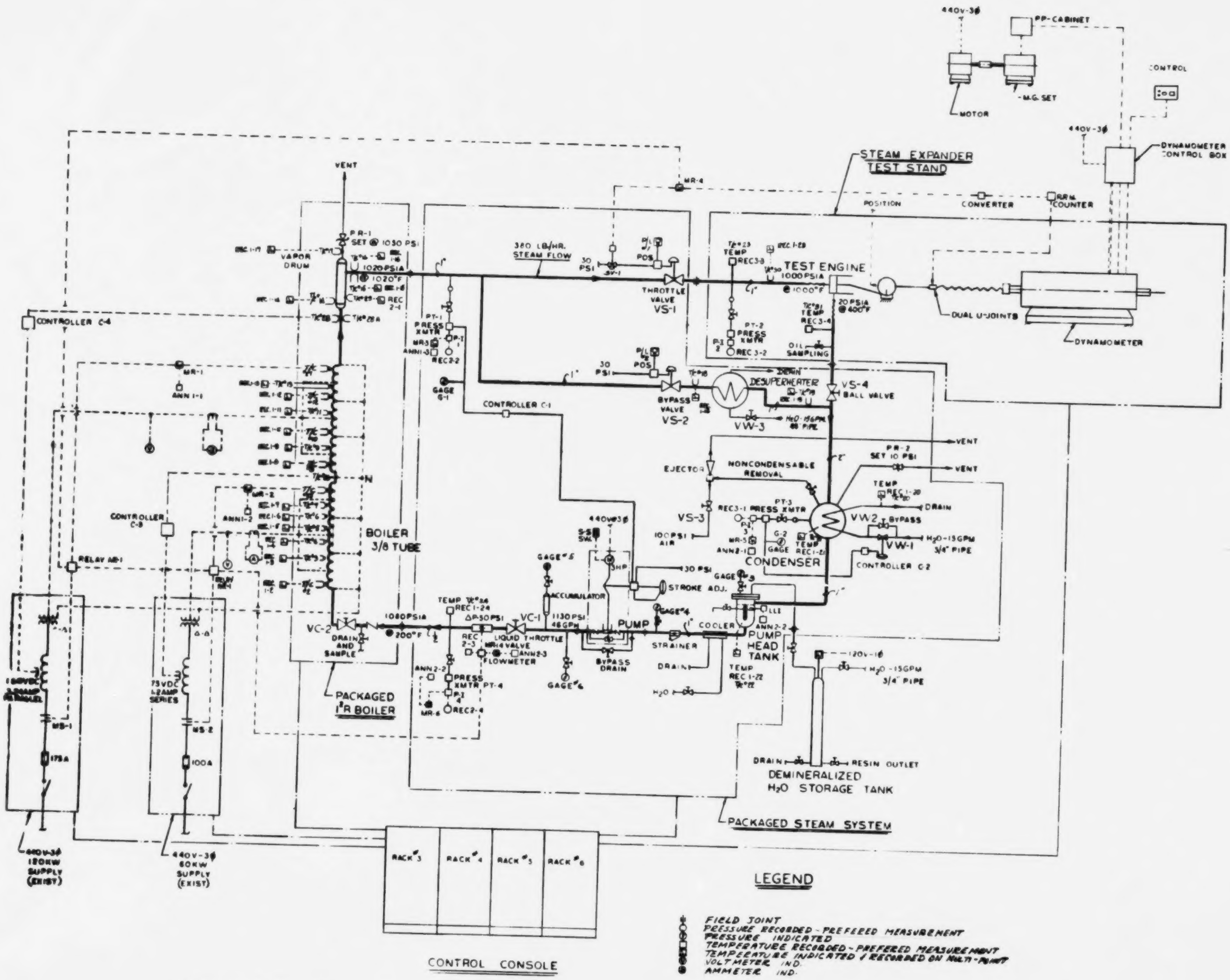
440V-3φ
PP-CABINET
MOTOR
M.G. SET
CONTROL
DYNAMOMETER CONTROL BOX
STEAM EXPANDER TEST STAND
POSITION
CONVERTER
R.P.M. COUNTER
DUAL U-JOINTS
DYNAMOMETER
TEST ENGINE
1000 PSIA
1000°F
20 PSIA
400°F
OIL SAMPLING
VENT
P.R-1 SET @ 1030 PSI
VAPOR DRUM
1020 PSIA
1020°F
380 LB/HR. STEAM FLOW
THROTTLE VALVE VS-1
MR-4
CONTROLLER C-4
PRESS XMTR
PT-1
PT-2
GAGE G-1
CONTROLLER C-1
BYPASS VALVE VS-2
DESUPERHEATER
VW-3
VS-4 BALL VALVE
PR-2 SET 10 PSI
VENT
EJECTOR
NONCONDENSABLE REMOVAL
VS-3
100 PSI AIR
PT-3 PRESS XMTR
CONDENSER
VW2
BYPASS
VW-1
DRAIN
H2O-15GPM 3/4" PIPE
CONTROLLER C-2
BOILER 3/8 TUBE
CONTROLLER C-3
RELAY AB-1
VC-2
DRAIN AND SAMPLE
1080 PSIA
200°F
ΔP-50 PSI
VC-1
LIQUID THROTTLE VALVE
FLOWMETER
1130 PSI 46 GPM
ACCUMULATOR
GAGE #5
GAGE #6
GAGE #4
GAGE #3
440V-3φ
30 PSI
STROKE ADJ.
PUMP
BYPASS DRAIN
STRAINER
COOLER
PUMP HEAD TANK
120V-1φ
H2O-15GPM 3/4" PIPE
DRAIN
RESIN OUTLET
DEMINERALIZED H2O STORAGE TANK
PACKAGED STEAM SYSTEM
PACKAGED I²R BOILER
PRESS XMTR PT-4
MR-6
REC2-4
180VDC 3.0AMP PARALLEL
MS-1
175A
73VDC 1.2AMP SERIES
MS-2
100A
440V-3φ 120KW SUPPLY (EXIST)
440V-3φ 60KW SUPPLY (EXIST)
RACK #3
RACK #4
RACK #5
RACK #6
CONTROL CONSOLE
LEGEND
FIELD JOINT
PRESSURE RECORDED - PREFERED MEASUREMENT
PRESSURE INDICATED
TEMPERATURE RECORDED - PREFERED MEASUREMENT
TEMPERATURE INDICATED & RECORDED ON MULTI-POINT
VOLTMETER IND.
AMMETER IND.

Figure 5-8. Steam Expander Test Facility Schematic (GE Dwg. 707E676).

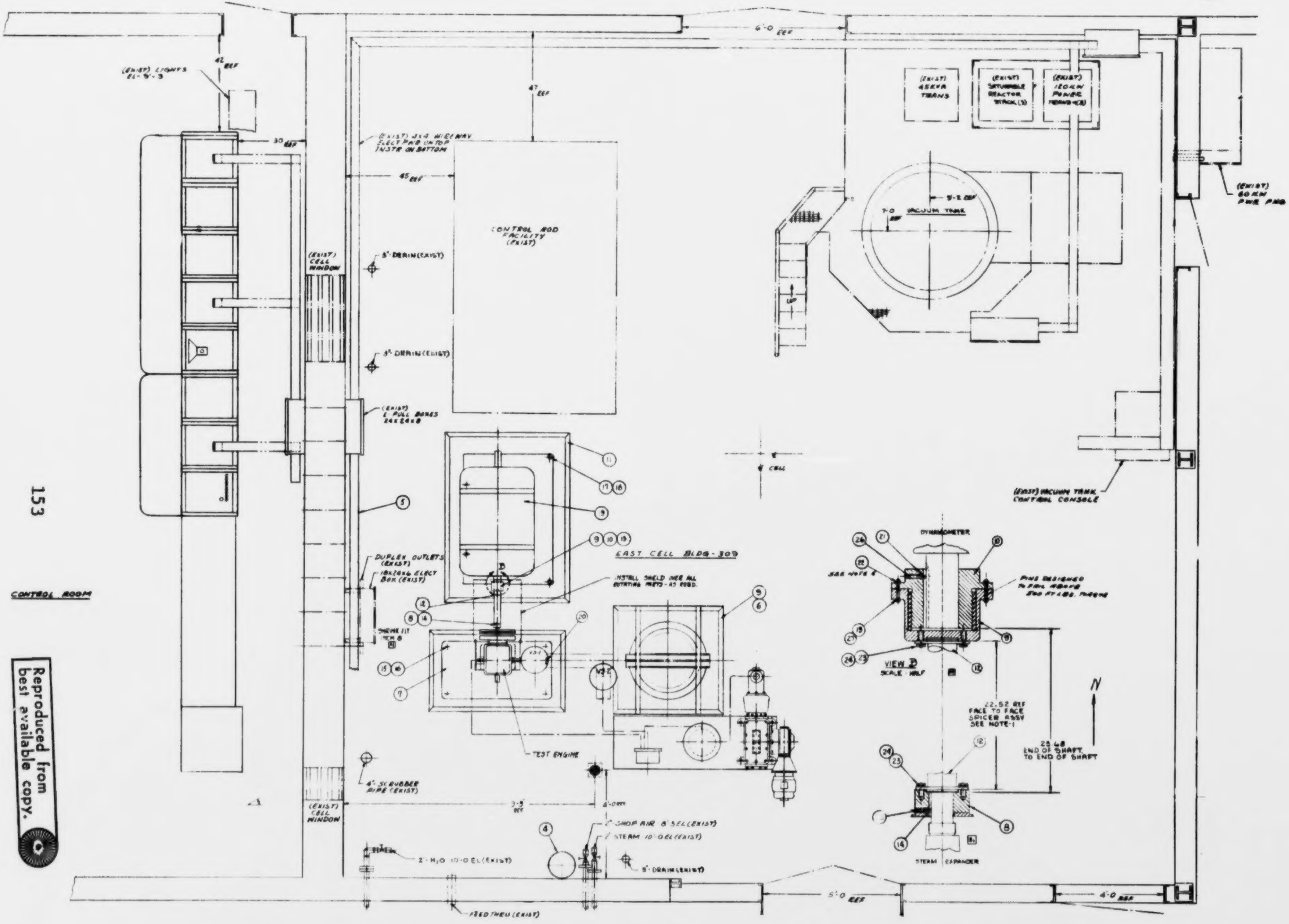
(EXIST) LIGHTS EL-9'-3
42 REF
30 REF
(EXIST) 4x4 WIREWAY ELECT PWR ON TOP INSTR ON BOTTOM
47 REF
45 REF
6'-0 REF
CONTROL ROD FACILITY (EXIST)
(EXIST) CELL WINDOW
5" DRAIN (EXIST)
3" DRAIN (EXIST)
(EXIST) 2 PULL BOXES 24x24x8
(EXIST) 45KVA TRANS
(EXIST) SATURABLE REACTOR STACK (3)
(EXIST) 120KW POWER TRANS (3)
(EXIST) 60KW PWR PNG
VACUUM TANK
7-0 REF
9'-2 REF
UP
℄ CELL
(EXIST) VACUUM TANK CONTROL CONSOLE
CONTROL ROOM
DUPLEX OUTLETS (EXIST)
18x24x6 ELECT BOX (EXIST)
SHRINK FIT ITEM 8
EAST CELL BLDG-309
INSTALL SHIELD OVER ALL ROTATING PARTS - AS REQD.
TEST ENGINE
4" SCRUBBER PIPE (EXIST)
(EXIST) CELL WINDOW
3-9 REF
4'-0 REF
2" SHOP AIR 8' 3 EL (EXIST)
2" STEAM 10'-0 EL (EXIST)
2-H₂O 10-0 EL (EXIST)
5" DRAIN (EXIST)
FEED THRU (EXIST)
5'-0 REF
4'-0 REF
DYNAMOMETER
SEE NOTE 6
PINS DESIGNED TO FAIL ABOVE 500 FT-LBS. TORQUE
VIEW D
SCALE - HALF
22.52 REF FACE TO FACE SPICER ASSY SEE NOTE-1
25.48 END OF SHAFT TO END OF SHAFT
N
STEAM EXPANDER
4
5
6
7
8
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Figure 5-9. General Arrangement of the Steam Expander Test Facility (GE Dwg. 47R199790).

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steam at a rate of 400 lb/hr. Three hundred and eight (308) feet of 3/8" OD x .031" wall, Type 316 stainless steel seamless tubing is coiled on a two (2) ft. diameter with a pitch of 1" to form the once-through boiler and superheater section. Calculated pressure drop through the steam generator at rated output is 80 psi. The coil terminates in a small vapor drum which discharges steam through 1" schedule 40 piping to a throttle valve and then to the expander. The steam generator is equipped with a pressure relief valve, and will be code stamped in accordance with the ASME Power Boiler Code. Design pressure is 1100 psi and design temperature is 1200°F. Two separate three phase SCR controlled power supplies will provide up to 180 KW of electrical power to the steam generator coil. Automatic and manual modes of power control will be provided.

A by-pass control valve and desuperheater permit checkout of the test facility prior to installation of the test expander. The by-pass valve and expander throttle valve will also permit inlet steam flow control to the expander with a constant boiler heating rate. This form of control is intended to be used during startup and shutdown of the expander.

A General Electric Company Model TLC-65/50 HP, 2500/750 RPM D.C. type dynamometer will be used to load (up to 65 HP) the steam expander, and it is capable of motoring (operating as a DC motor) up to 50 HP. Speed regulation of ± 1% of rated speed can be maintained during both modes of operation. The dynamometer is capable of delivering 500 ft. lbs. of torque for one minute and it will be used to rotate the expander until the expander becomes self sustaining.

The steam expander will be coupled to the dynamometer via a spicer shaft utilizing universal joints at each end and a splined telescoping center section. A shear pin adapter at the dynamometer shaft connection will limit inertial torque loading of the expander by the massive dynamometer in the event of "freeze-up" or other failure which might suddenly stop the expander. The dynamometer is mounted on a separate foundation from the expander and is isolated from the floor of the test cell for minimum vibration. An electronic (strain gage type load cell) system will be used for torque measurement.

The expander exhaust is piped to a water cooled condenser which is capable of handling the 400 lb/hr of steam, and which can maintain 20 psia discharge pressure. Steam discharging from the desuperheater also enters the condenser during by-pass operation.

A water storage tank equipped with liquid level indicator, filling system and oil removal (overflow blown-down) system receives the condensate from the condenser. Demineralized makeup water will be supplied to the tank directly from the demineralizer as required during operation.

A piston type motor driven pump with automatic stroke adjustment feeds the boiler with water. A small heat exchanger ahead of the pump provides cooling to prevent pump cavitation. The pump is capable of providing up to 1.0 gpm at 1200 psi to the boiler. A built-in pressure relief valve prevents an overpressure of the pump. Pump discharge pressure and flowrate will be controlled automatically by sensing boiler discharge pressure. Steam flowrate will be determined by measuring boiler feed water flow. A turbine type meter is provided for water flow measurement.

A control console located in the control room will contain all the operating controls and instrumentation for both the facility and steam expander. All critical parameters will be recorded. Protective circuits for overspeed, overtemperature, and overpressure are provided. Figure 5-8 indicates the position of temperature and pressure sensors throughout the facility, and parameters which will be automatically annunciated when out of limits. Temperature, pressure or expander speed above set limits will result in an automatic shutdown of the complete facility.

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